LUBRICATION ENGINEERING

RESEARCH APPLICATION EDUCATION MAINTENANCE

Friction Machine (see p. 83)

JOURNAL OF THE AMERICAN SOCIETY OF LUBRICATION ENGINEERS

FARVAL—*Studies in Centralized Lubrication*
No. 218

Farval lubrication first choice for Hill-Acme Upsetter

On this 1¼-inch Hill-Acme Upsetter, Farval's timing cycle can easily be changed to conform with the service of the macnine. Amount of lubricant delivered to each bearing is also adjustable.

BUILDING peak performance into their machines is a point-of-pride with Hill-Acme. That's why you'll find Farval lubrication systems on H-A machines such as this upsetter.

When the going's rugged, with shock, vibration and wear trying to take their toll, Farval continuously guards 59 vital bearings . . . continuously delivers measured amounts of lubricant at regular intervals.

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Wherever you see the sign of Farval—familiar valve manifolds, dual lubricant lines and central pumping station—you know a machine is being properly lubricated.

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LUBRICATION ENGINEERING

RESEARCH APPLICATION EDUCATION MAINTENANCE

Volume 14, No. 3

March, 1958

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LUBRICATION IN THE NEWS

New Energy Sources May Help Eliminate Present Power Losses

One hundred years from today, there will be three times as many people in the world as there are now.

With this projected population increase, annual world energy demand will go up between 10 and 100 times. Fossil fuel production (coal, oil, gas) in the United States will fall short of demand within 10 to 30 years. These predictions place an "urgent" stamp on new sources and methods of creating electrical energy.

According to Dr. Morris A. Steinberg, head of the metallurgy department of Horizons Incorporated, "The whole problem revolves around the highly inefficient way in which electrical energy is produced today. A turbine, powered by either steam or water, spins a generator which produces electricity. By the time the electrical flow reaches a transmission line, its delivered kilowatt load is only about 30 per cent of what the original fuel material *could* have delivered.

"If this efficiency level could be brought up to 60 to 80 per cent, our fossil fuel resources would last a great deal longer."

Careful examination is being given by scientists to the possibilities of generation of electricity, creating current without the use of moving parts. Two major areas of study are under investigation: (1) Solar or nuclear power sources for direct generation of electrical energy and (2) generating electricity in electro-chemical cells. (This second is the more immediately promising of the two.)

One fuel cell, using a reaction of hydrogen and oxygen, is now in limited use on special applications. The cell, similar to a battery, has porous electrodes and does not degrade. In operation, hydrogen is diffused through the positive electrode, reaching with hydroxyl ions, formed in the electrolyte by diffusion of oxygen through the negative electrode. The overall reaction is the formation of water plus two electrons. Theoretical voltage of this reaction is 1.01 volts. Life expectancy of the cell is unlimited; neither electrolyte nor electrodes are consumed during operation.

Despite the advantages of such power generation, several problems exist before these devices can come into popular use. The biggest problem is cost—due to the lack of low-cost hydrogen. Another dilemma is that a fuel cell produces direct current—most homes and commercial establishments now use alternating current. Converting direct current into alternating current results in the same loss in efficiency.

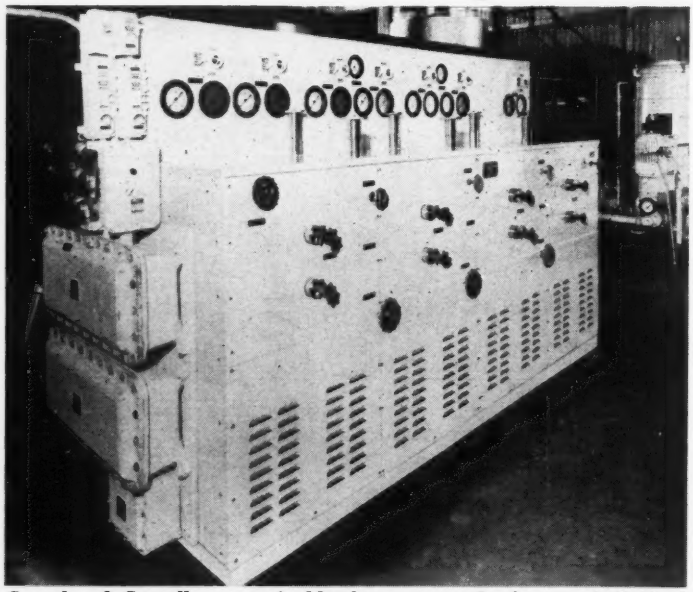
However, there are strong indications that, within a few years, fuel cells will be in common use on special purpose applications, particularly where silent portable power sources are required.

Infrared Rays Seen As Key To Proper Oil Identification

Analytical chemistry and statistics have joined forces in the battle against incorrect branding of motor oils. A new method, developed by Armour Research Foundation of the Illinois Institute of Technology under sponsorship of the Pennsylvania Grade Crude Oil Assn., makes it possible for the first time to determine differences between virgin and reclaimed oils.

Until now, no consistent differences in chemical properties of the two oils had been found to permit development of an identifying procedure. In a new approach to the problem, Foundation scientists have shot infrared rays through the oils and have found sufficient statistical differences to provide identification of one from the other. The method detects the multiplicity of crude sources in the reclaimed oil in contrast with the single or small number of crude sources of the virgin oil.

First Fully Automatic System For Additive Blending Installed

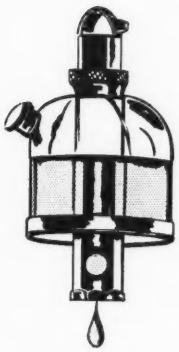


Console of Cornell automatic blender, composed of seven individual panels through which Oronite lubricating oil additive components are simultaneously blended at accurately controlled proportions.

An important step forward in the production of lubricating oil additivies has been put into operation by Oronite Chemical Company with the introduction of the world's first fully automatic system of additive blending. A marked feature of this new concept in additive manufacturing is the ability of accurate, high-speed blending of as many as seven different additive components in one automatic operation. Oronite officials claim that the new system answers the need for economically-priced, highly complex additive formulations that are consistently uniform. Lubricating oils can now be compounded to match precise customer requirements at any quality level.

Heart of the new operation at Oak Point, La., is a high-speed blending machine which carefully proportions each additive component into the desired additive formula. Capacity is 90 gallons of additive per minute; a 10,000 gallon tank car can be filled in approximately two hours.

The Sight Glass



Highlights of Articles Scheduled for Coming Issues!

Design and Testing Considerations of Lubricants for Gear Applications

With the stress toward higher speeds, more load, and less weight, the higher capacity gearing trends of modern industry have placed an unusual demand upon available gear lubricants. Many gear designs are so dependent upon the lubricant that the lubricant has become a constructional gear material. The types of gear failures related to the lubricant, wear and scoring, have been investigated and some of the tests and procedures have been used to evaluate an oil for gear applications.

Design and Operation of Ball Bearings for use in Pressurized-Water Reactor Systems

Early in the development of nuclear power reactors, commercial ball bearing materials and designs were found unacceptable for service in high temperature, high purity, pressurized-water power reactors. Commercial bearings seized, galled, and wore severely, even under small fractions of their rated load for lubricated service. After searching for new materials suitable for use in reactors fabricating and testing the new materials with conventional design concepts, it became apparent that basic bearing designs must be modified to attain required improvement in bearing wear and corrosion resistance. The design approach to this problem is described.

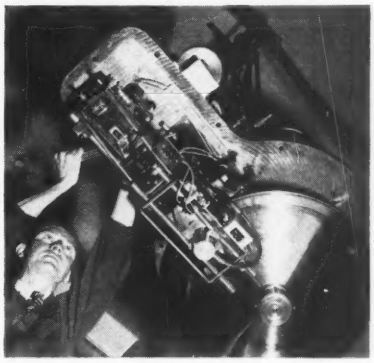
Molybdenum Disulfide and Related Solid Lubricants

Wear studies of bonded films of solid lubricants show variations in wear life and friction dependent upon type of lubricant and application. Molybdenum disulfide, graphite, and boron nitride are compared as bonded coatings. Wear life of organic bonded films are found to be affected by many impurities and prolonged heat exposure. Friction and endurance data obtained using metals as bonding agents indicate that, with further development, inorganic metal binders will give coatings that surpass resin binders.

Increased Machine Productivity Demands Well Designed Lubrication Practices

Automation of machine tools has compounded the problems of the lubrication engineer since down-time means loss of complete units rather than a single operation. Transfer, or specially designed equipment contains lubrication systems whose cost is itemized as an accessory, instead of being allocated by the tool builder and determined by competitive practices. The efficient application of oil or grease is thereby very forcibly called to the attention of management. By understanding and applying his specialized knowledge, the lubrication engineer can suggest to the machine tool builders the latest engineering designs available from the manufacturer of lubricating devices for incorporation into the specialized machinery required for automated manufacturing methods.

COVER STORY



The high speed friction and wear machine shown in the cover photo was designed by the staff of The Franklin Institute Laboratories, Philadelphia, for determining the friction and wear characteristics of materials undergoing dry sliding friction at high speeds and pressures. It is one of a series of special friction research machines built in the laboratories; subsequent additions to the series are a water-lubricated-bearing test machine, journal bearing test machine, and devices for gas-lubrication studies.

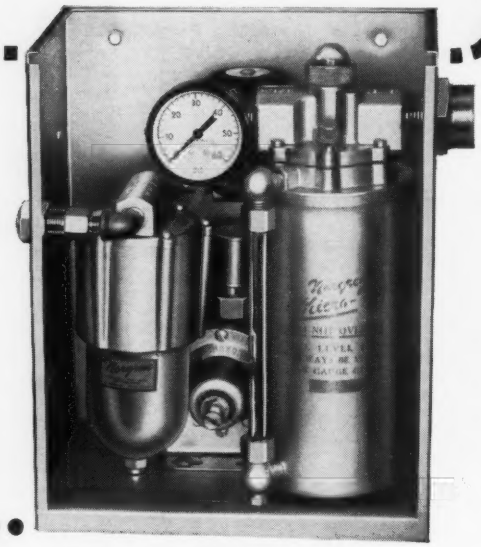
Originally designed for studies of wear in gun barrels, the machine has since provided valuable data for many research programs. A program is now under way to study the generation of heat at the interface, dissipation of the heat, and the accompanying effects on friction and wear.

With the disc rotating at speeds up to 20,000 rpm, a loading mechanism presses the specimen (usually a small slug about 1/16 inch in diameter) against a flat annular ring on the rim of the disc, simultaneously moving it across the ring so that the sample traces a spiral path over a continuously fresh surface for as much as 40 feet. Maximum load is 120 lb., giving pressures up to 50,000 psi. Load and frictional force are monitored continuously by electric strain gages and recorded by both a multichannel oscilloscope and a 35-mm drum camera.

Tests normally last for from 0.01 second to a few seconds; for tests as long as six hours, another loading mechanism is available and a multi-point chart recorder accompanies it. These longer tests, of course, are made so that the specimen retraces its path.

Other projects engaging the skills of the Friction and Lubrication Section of the laboratories include the design of hydrodynamic and hydrostatic bearings for many unusual applications; some — such as the bearings for the new 140-foot radio telescope — requiring modern mathematical techniques for solution (e.g., the electric analog). The section is also carrying out a long-range research program on gas-lubricated bearings for the Office of Naval Research.

Norgren MICRO-FOG LUBRICATION

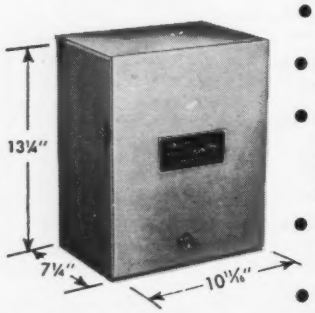


NEW...

Compact, Tamper-Proof Cabinet Units

lubricate all bearings, gears, chains and cams on a machine

These new Norgren units provide ideal lubrication by applying just the right amount of air-borne oil fog to all bearing surfaces. The standard unit includes an automatic-drain air line filter, a solenoid valve, a pressure regulator, a pressure switch and a MICRO-FOG Lubricator... *and provides the following important advantages:*



1 QT. OR 2 QT. OIL CAPACITY

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- **TAMPER-PROOF**—Totally enclosed key-locked cabinet prevents tampering.
- **SPLASH-PROOF**—Coolant and corrosive liquids cannot reach interior of enclosure.
- **CONVENIENT TO SERVICE**—Removable one-piece top and side panel—secured by key lock—provides easy access for authorized personnel. Visual oil feed is proof of lubrication. Oil level gauge glass provided on lubricator.
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- **HEATER AND OIL LEVEL CONTROL OPTIONAL**—Thermostatically-controlled Heater, for 1 Qt. units, maintains lubricant temperature and viscosity. Oil level control actuates alarm or stops machine when oil needs replenishing.

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If it's Norgren... It's Dependable.

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PERSONALS

On January 1, 1958 Brooks Oil Co.'s sales and engineering offices were moved from Pittsburgh to Cleveland. The Pittsburgh office will continue as a district sales office.

Stuart S. Carlton has been appointed field sales representative for the northeastern United States by Horizons Inc., Cleveland process and materials research organization.

Dean of conveyor engineers Myron A. "Mike" Kendall, Stephens-Adamson Mfg. Co. board chairman, was awarded the American Society of Mechanical Engineers' "Fellow Award."

Myron Kendall

Dr. Meyer

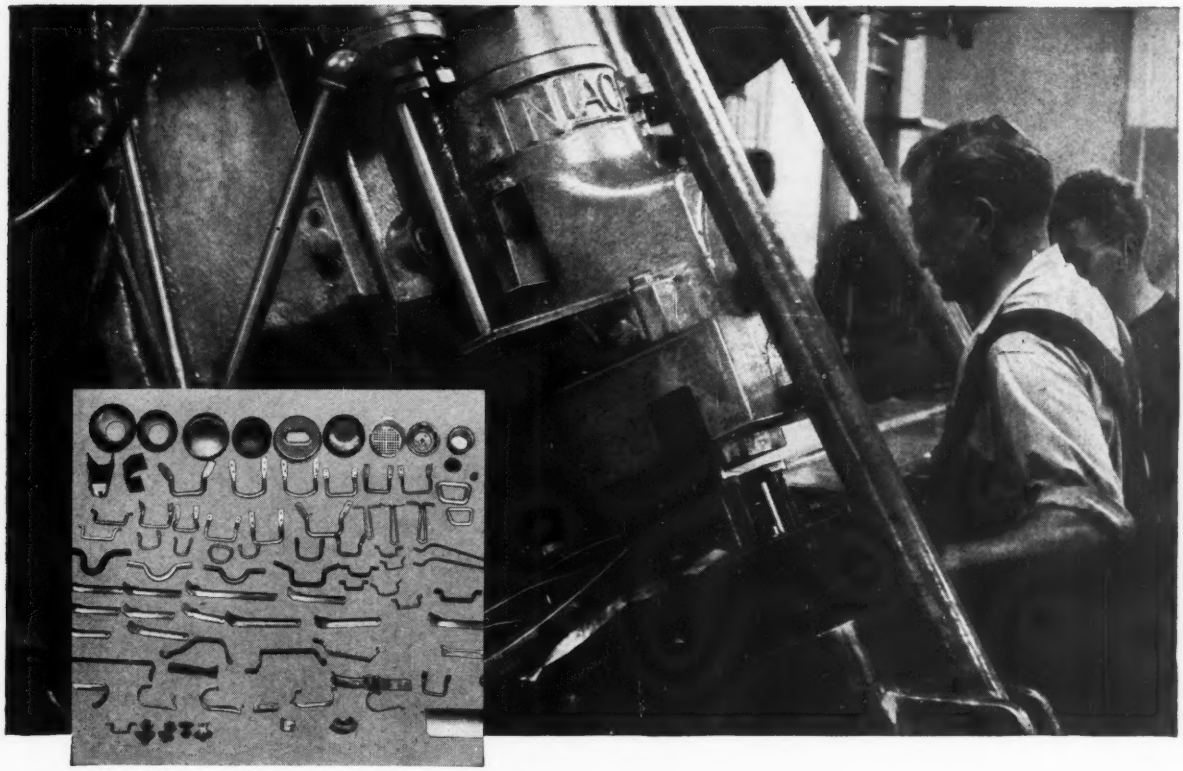
Trabon Engineering Corp., Solon, Ohio, has announced the appointment of the J. N. Fauver Co. of Cincinnati as exclusive distributors in the southern Ohio, Indiana, western Kentucky and Ohio River Valley areas for Trabon centralized lubrication systems.

The appointment of Dr. Erich Meyer as vice president of L. Sonneborn Sons, Inc. has been announced. Dr. Meyer moved up from the position of director of the department of industrial research. He will continue to be responsible for Sonneborn's industrial research, acting as liaison between research, manufacturing and sales.

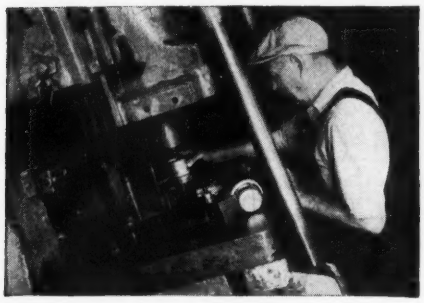
Joy Mfg. Co. of Pittsburgh and the Industrial Filtration Co. of Lebanon, Ind. have announced a franchise agreement whereby Joy becomes exclusive distributor for the Delpark Filter to the mining industry in Canada, Mexico and the U. S. (except California, Oregon, Washington, Nevada, Idaho, Arizona, Alaska and Hawaii). Joy also has the exclusive franchise to sell the filter to any other industry in the same area when it is sold as a component part of the Joy-microdyne dust collector.

Herbert M. Schneider has been named coordination engineer for manufacturing and production engineering for Acheson Industries, Inc.

Robert C. Gray has been appointed assistant marketing manager of Orr & Sembower, manufacturers of packaged automatic heat exchange equipment in Reading, Pa.



"We stamp them all using just 2 Cities Service Oils!"



Presses shift to numerous operations without ever changing oil, thanks to versatility of the 2 Cities Service oils used for all jobs.

Handles by the thousands are produced by Res for cooking utensils and similar items. Finish is excellent, thanks to Cities Service oil.

...says Res Manufacturing Company Milwaukee, Wisconsin

It's doubtful if anyone makes a more diversified line of stamped and formed metal products than Res Manufacturing Company.

With its 21 presses, ranging from 15 to 200 tons capacity, Res turns out millions of articles a year . . . wire handles, metal handles, wire forms, drawn shells, lifting loops, and parts for electrical controls, to mention just a few.

"All require the best possible finish and all get it with the use of just two Cities Service oils," says Assistant Plant Manager Herbert W. Krueger.

"For drawing, shaping, and punching aluminum, we use Cities Service Chillo #2 with excellent results. Where drawing, swedging or forming of steel and brass is required, we use Cities Service Chillo 12 and get equally fine results.

"Not only do these two Cities Service oils handle all our jobs and produce the best possible finish, but we also find they increase tool life and hold scrap to a minimum. In short, these are the best oils we have ever seen."

Like Res Manufacturing, chances are your operation can be simplified with the right Cities Service oils. Talk with a Cities Service Lubrication Engineer. Or write: Cities Service Oil Company, Sixty Wall Tower, New York 5, N. Y.

CITIES SERVICE

QUALITY PETROLEUM PRODUCTS

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CHECKS "STICK-SLIP" ON A GRAND SCALE

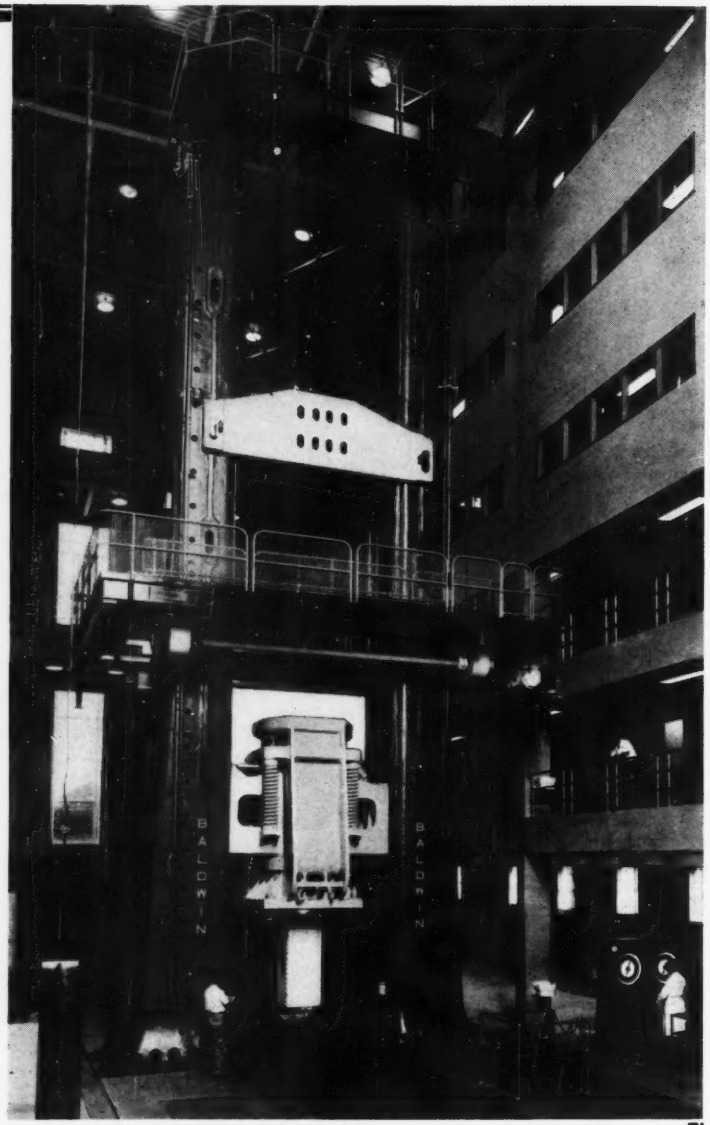
When engineers of Baldwin-Lima-Hamilton Corporation designed and built this mammoth 5,000,000 pound universal testing machine for Lehigh University, they realized from years of experience that extreme testing pressures could cause destructive "stick-slip" action in the test specimen grips and the loading screws.

That is why MOLYKOTE Lubricant was generously applied to both the grips and the screws before the first trial specimen was broken in this giant universal tester.

Baldwin engineers, as well as engineers throughout industry, have come to depend upon MOLYKOTE Lubricant in extreme pressure applications.

MOLYKOTE is manufactured from the purest molybdenum disulfide powder available to industry today. It is compounded under laboratory supervision to retain this purity in the manufacturing process and has achieved an enviable reputation for overcoming the toughest lubrication problems in industry.

MOLYKOTE Lubricant is distributed throughout the world, wherever industry faces the obstacles of providing adequate lubrication in extreme pressure and extreme temperature applications.



5,000,000 pound B-T-E Universal Testing Machine installed in Fritz Engineering Laboratory, Lehigh University, Bethlehem, Pa. Built by Baldwin-Lima-Hamilton Corporation, Eddystone, Pa.

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Lube Lines

by A. F. Brewer*

PAST HISTORY

"History repeats itself" is a cliche which is often more truthful than poetic. The history of the world is replete with significant examples. Were we able to compile a history of plant lubrication covering a broad range of industries, undoubtedly there also would be found some glaring examples of history repeating itself. Some of these could well be regarded as warnings. Today the lubrication engineer profits from past experience. Where it has been favorable, both he and his company are the beneficiaries. Where it has involved "headaches," he is forewarned so that whatever the cause, he plans so that it won't happen again. In other words past history as far as lubrication is concerned contributes advantages and profits.

The Advantages

(a) Records of machine performance.
(b) Time schedules or periods of operation between re-lubrication or overhaul.
(c) Performance ability of lubricants and lubricating devices.

Who Profits

(a) The lubrication engineer.
(b) The maintenance personnel.
(c) The purchasing department.
(d) Inventory and handling procedures.
(e) Management, by virtue of reduced costs of lubrication and maintenance.
(f) Production — more dependable.

Records of machine performance are directly related to effective lubrication. In terms of production output they are invaluable in determining the unit cost of lubrication and maintenance and in developing standards which can be used to good advantage when any change in method or means of lubrication, maintenance procedure or operating routine is contemplated.

Regularity is beneficial in the care and maintenance of any piece of machinery. The most homey example involves one's automobile where periodic motor oil change and chassis lubrication are so conducive to dependable transportation. Scheduling periods of re-lubrication, machine inspection and overhaul procedure are equally as important in the operation of industrial and power plant machinery. Where the lubricating system can be set for timed or measured delivery or circulation of grease or oil, the personal responsibility of the operator is confined to inspection and regulation of delivery mechanisms. Records of machine performance as to production, lubricant consumption and maintenance costs fit right in with the establishment of such schedules.

Records of this nature also become a criterion as to the suitability of lubricants or the efficiency of means of application. "Comparisons may be odious" where personalities are involved, but comparison of recorded data as to lubricant consumption and cost of maintenance can contribute markedly to the kind of operation which pays off in dividends instead of unwarranted time loss and need for costly repair parts.

In a period of reduced inventories when it may be necessary to take some units out of service, it becomes all the more important to protect the operating plant machinery against excessive wear and unwarranted replacement of working parts. Under such conditions, the performance value of lubricants and the ability of the lubrication engineer to evaluate this performance factor must not be overlooked by management. Past history in certain industries might have indicated that "cannibalism" of idle machinery was acceptable. But no machine is ready to go when certain of its vital parts have been removed for service on more fortunate operating units. It's tough enough to have to pay interest on an investment which is not producing, but to pay interest on a machine which *cannot* produce is just not good business. So don't let history repeat itself. To take a partly dismantled machine out of mothballs and put it into operating condition again could involve an expense that would far exceed any temporary inventory savings. Remember also that parts last longer and function more dependably when properly lubricated. We do not need to refer to history to prove this; it is axiomatic.

*Consultant, and author of **Basic Lubrication Practice**, see p. 47 Feb. L. E.

ASLE News Notebook

Compiled by
R. D. McCormick

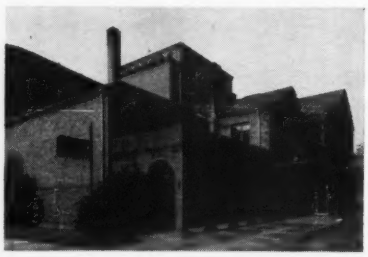
Shown are members of the Cleveland Convention Planning Committee. Front row: G. L. Smith, C. Harrod, J. Hurley, E. Gesedorf, P. A. Asseff and E. Halligan. Rear: W. E. Campbell, L. O. Witzenberg, E. Baumgardner, and T. Osters.

Cleveland Section Completes '58 Annual Meeting Plans

Planning for the various non-technical functions for the forthcoming annual meeting of the Society, Cleveland, Ohio, April 22-24, are key members of the Cleveland host section. Members of the 1958 Convention Committee of the ASLE are: Ed Gesdorf, Chairman, 1958 Convention Committee; George L. Smith, Secretary; Carl Harrod, Chairman, Convention Finance Committee; Gene Halligan, Chairman, Publicity & Advertising Committee; Bill Campbell, Chairman, Welcoming Committee; J. Hurley, Vice-Chairman Convention Committee; T. Walsh, Chairman, Sessions Coordinating Committee; Ed Baumgardner, Chairman, Exhibits Committee, and P. A. Asseff, Chairman, Entertainment Committee.

A special welcoming party is planned by the Cleveland section for Monday, April 21, from 5-7 P.M. at the Cleveland Engineering Society building. Admission to the cocktail party will be by membership card or registration badge. Registration badges will be available to members and guests at the Hotel Cleveland from the advance registration desk.

Immediately following the advance registration at the Hotel Cleveland, Monday afternoon, April 21, 1958, the Cleveland host section will sponsor a cocktail party at the Cleveland Engineering Society's auditorium. Start of the welcoming activities are scheduled for 5 P.M. Admission will be by membership card or advance registration badge only. An admission fee of $1.50 will be charged in lieu of registration proof. Members and guests planning to attend the 1958 Annual Meeting are urged to arrive at the Hotel Cleveland in time to register *before* the cocktail party.



The Cleveland Play House, a non-profit professional resident theatre, which staffs and keeps three separate theaters going with current and classic drama.

Ladies' Program Features TV Tour

Ladies attending the 1958 Annual Meeting will be able to attend the *Dorothy Fuldheim Show*, televised from Cleveland's station WEWS. Luncheon and a tour of the studios of WEWS will precede the show scheduled for 1-2 P.M. Afterwards, the ladies' program committee has scheduled a tour of Nela Park. Tickets to a play at the Play House (see photo) will be supplied for Tuesday evening. Wednesday's program includes tours of the city and luncheon at one of the universities. In compliance with past requests, no formal program is planned for the last day of the Annual Meeting. Maps, guide books and sightseeing arrangements will be provided upon request, but the last day has been set aside to permit shopping trips in the Cleveland area.

Mrs. Campbell

Miss Fuldheim

Plans for the ladies' program are in the capable hands of Mrs. W. E. Campbell. Assisting Mrs. Campbell are Mrs. E. Crankshaw, Mrs. L. O. Witzenburg, and Mrs. R. H. Josephson.

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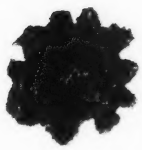
EFFICIENCY EXCELLENCE

BARAGEL
Provides manufacturing ECONOMY resulting in lower grease cost.

BARAGEL



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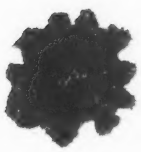

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Great Names in Steel Making #VIII

OSCAR L. MAAG, *recipient of the ASLE National Award in 1956, was preeminently a lubrication engineer. Yet every mill in America that operates on roller bearings testifies to Oscar Maag's contribution to modern steel making.*

As sometimes happens in the initial trial of a new process or material, the first test of roller bearings in a rolling mill was quite successful. But as soon as increased speed and pressure were attempted, in a second installation, serious trouble developed.

The big Timken bearings (they measured 30"x16" in diameter) were taken back to the company's plant at Canton, Ohio, where they were lapped and studied on the test stand. The cause of failure was in the calcium soap grease used to lubricate the bearings. Furthermore, it was evident that no type of lubricant then available was capable of handling the load.

At that critical moment Oscar Maag, Timken's lubrication specialist, had an inspiration. Experimentally, he added sulphur chlorinated base cutting oil to the grease.

It was the first E. P. (extreme pressure) lubricant applied to rolling mill bearings, and it worked perfectly. Not only did it solve the immediate problem—it cleared the way for development of roller bearing mills in which speeds exceeding a mile-a-minute and pressures to 25,000 psi are successfully handled by today's Ironsides "E.P." lubricants.

As specialist for 65 years in the manufacture of lubricants for steel making, The Ironsides Company is especially pleased to honor Oscar Maag of The Timken Roller Bearing Company for his great contributions to our industry. It was our privilege, here at Ironsides, to work closely with Oscar Maag as our Company's consultant following his retirement from Timken, and to know him as both a great technologist and good friend. The Ironsides Company, Columbus 16, Ohio.

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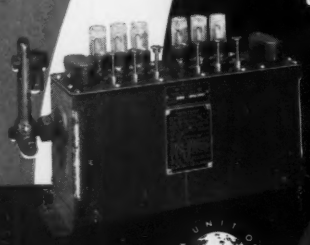

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A new compound made from fluorinated silicone rubber reportedly has excellent high and low temperature fuel and oil resistance. Intended for use as "0" rings and other pressure type seals where some conventional fluorinated silicone elastomers are said to be unsuccessful because of low modulus, the new 70 SDH compound is claimed to resist extrusion into clearances and be resistant to a variety of hard-to-seal fluids, air and fuel and air mixtures from —80° to 400°F. A previously difficult sealing problem that the compound handles is the sealing of silicone oils and greases throughout an extreme temperature range. For additional information on TH 1057, write to G. W. Van Cleve, Vice Pres.-Sales, Stillman Rubber Co., 5811 Marilyn Ave., Culver City, Calif.

KINEMATIC VISCOSITY BATH

A new, complete Temp-Trol kinematic viscosity bath which provides ±0.02°F. uniformity has been announced. This new unit reportedly allows accurate kinematic viscosity determinations of Newtonian liquids at minimum cost. The following advantages are claimed: (1) high degree of uniformity (2) maximum utility (3) conformance with top standards (4) broad applicability—accomodates any modified Ostwald, Ubbelohde, or S.I.L. suspended level viscometer (5) wide range measurement—including viscosities of 0.4 to 16,000 centistokes (6) broad operational capacity and (7) simplified maintenance. Further information can be obtained from Precision Scientific Co., 3737 W. Cortland St., Chicago, Ill.

PRECISION BALANCER

A light-weight low-cost, precision electronic industrial balancer with weight and size ranges for specific use in automobile and truck engine rebuilding, plant maintenance, quality control and short-run production lines has been developed. The unit is capable of balancing to .004 inch-ounces all rotating parts weighing between ¼ lb and 300 lb with diametral range of from ¼ in. to 30 in. The third model in a series of industrial balancers marketed by the manufacturer, the new unit fills the gap between two heavier balancers with weight limits to 1,000 and 5,000 lb and diameters from one in. to 68 in. Details on Model 702 are available from the Stewart-Warner Corp., 1826 Diversey Pkwy., Chicago 14, Ill.

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American Soc. of Lubrication Engineers, Publisher

84 E. Randolph St.

CHICAGO 1, ILLINOIS

American Society of Lubrication Engineers
84 E. Randolph St., Chicago 1, Ill., U. S. A.

Gentlemen:

Please submit my name to the chairman of the ASLE Industry Council, as checked, as a prospective member of this council.

☐ Aviation ☐ Machine Tools ☐ Nonferrous Metals ☐ Railroads ☐ Steel

I am interested in the formation of the following councils: (specify)

☐_____ ☐_____

Please print

Name _____

Preferred address _____

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LUBRICATION ENGINEERING

RESEARCH APPLICATION EDUCATION MAINTENANCE

Journal of the American Society of Lubrication Engineers *Vol. 14 No. 3 March 1958*

Establishment of Industry Councils — Another Step Forward

Incorporation in ASLE of Industry Councils is now complete. Formal rules for their organization and operation were adopted at the last meeting of your Board of Directors following almost a year of planning and evolution of Industry Council work.

It is curious that so long a time has elapsed in the short ASLE history before this step was taken. Always aiming to smooth the turning of the wheels of industry, ASLE nevertheless concentrated its early attention on the individual technical aspects of our lubrication field. In so doing, our meetings and publications have for some years been a recognized clearinghouse for revealing the latest advances in bearings, gears, lubrication equipment, lubricants, and seals. Committees of the foremost workers in these activities have strongly promoted and accelerated this aspect of lubrication work.

The way is now open for a similar focus on lubrication on an industry-by-industry basis. In doing so, a lubrication group consisting of representatives from a single industry can simultaneously consider their problems with lubricant selection, scheduling, design of moving parts, and economics. This is already true for Councils in the following industries:

Aviation
Machine Tool
Nonferrous Metals
Railroads
Steel

It is planned that other Councils will be established from time to time as you find need for them.

The industry councils got their start with the growth of our fine National Railroad Lubrication Council. As the NRLC initially developed, there actually was found to be no place in the ASLE for its functions. Only through the guidance of our Organization and Operations Committee with the NRLC was your Board of Directors able to successfully set its preliminary plans for Industry Councils at the 1957 Annual Meeting. The Councils are now firmly placed in our Society with a parallel status to our Technical Committees, with coordination through a Joint Industry Council, and with direct representation at Board of Directors meetings by an Industry Coordinator.

The future of the Industry Councils is yours. Only through your active participation can their role in the development of American industry be fully achieved. As a start, indicate your interest on the attached post card, or simply drop a letter to the National Office.

E. R. Booser, *Chairman*
Organization and Operations Committee

Standardization of Plant Lubricants and Lubrication Practices

R. O. Kageff, Ford Motor Company, Dearborn, Michigan

In large companies the dollar volume spent on lubricants warrants simplification and standardization. Such a program designed to provide proper lubrication at a substantial savings has been initiated by the author's company. It involves education of lubrication personnel, a preventive maintenance program, automatic methods of lubricant application, and lubricant specifications.

In order to realize maximum results from such a program, it is necessary to designate a direct or functional control over all company operations. Controls must be on a continuous basis. Periodic surveys must be conducted to insure continued progress.

Lubrication personnel must be given the latest information on methods of lubricant application. They must be given instructions in modern methods of handling and storing lubricants. They must be informed of the importance of their work.

Preventive maintenance is a vital role in any lubrication program. To prevent lubrication failures is more important than correcting the cause of a failure. Such a program also indicates or pinpoints failures other than lubrication.

In this era of automation application, constant efforts to supplant the labor and detail of hand applications have resulted in more and increasing use of automatic lubricant systems. Not only have such methods proven economic, but they have provided a greater assurance that the equipment has been lubricated.

The purchase of lubricants by specification insures competitive prices. There are, of course, advantages and disadvantages in this method. One requirement is that lubricant specifications must be revised periodically to reflect the continuing improvement of products by lubricant suppliers.

There are no hard and fast rules for accomplishing a good lubrication program. The important consideration is that a lubrication program can be accomplished and the savings realized worth the work and time expended.

In large companies, the dollar volume spent on plant lubricants alone warrants standardization and simplification. In many such companies the annual cost for plant lubricants is figured in millions of dollars. While this expenditure may seem large, in most instances it actually reflects less than 1 percent of total manufacturing costs. However, this money carefully controlled and wisely spent, can reduce much larger costs for equipment repair and consequent production losses.

Such a program designed to provide proper lubrication should encompass all phases of lubrication from purchasing to consumption. Essentially, this would mean coordination of all activities concerned, that is, *Purchasing, Plant Engineering, Manufacturing,* and *Development.* This coordination may be accomplished by an individual, a department, or a lubrication committee depending on the size of the company.

In our Company, the Process Development Department functionally supervises all operations in the field of materials and processes. All the information available from suppliers, technical societies, and research institutes passes through this department. This information is read carefully for all new ideas and old ideas with new applications. While we are engaged in reviewing all types of materials, both production and nonproduction, lubricants are given considerable attention.

The information gained in this manner is then put into a more direct and practical form for distribution to all interested personnel. This form can be in the nature of recommended practices charts, processes, or material specifications. In this way, we are able to reach many of our plant personnel.

This information is also available to our maintenance personnel through a Lubrication Standards Manual published by our Central Staff Maintenance Department.

TRAINING COURSES

Our Staff Training Department contributes training courses for oilers and lubrication foremen. These courses cover the elementary principles of lubrication, the importance of lubrication work, methods of handling and dispensing lubricants, and lubrication equipment. In the case of the latter, a special comprehensive course is given on all centralized lubrication systems.

While these courses have helped immensely in equipping lubrication personnel with the knowledge necessary to perform their task, it must be remembered that proper attitude plays an important part in a good lubrication program. Your efforts as lubrication engineers to instill the proper degree of importance of the lubrication program in your lubrication personnel is reflected in their attitude towards their work. Lubrication personnel's attitude toward their work is usually in direct relationship to the attitude of management. While you may be well aware of how important lubrication programs can be, they are of no value unless the

Presented at the 12th Annual Meeting of the ASLE, April 15-17, 1957, Detroit, Michigan.

people working with you have the same awareness. Get this message across to the man dispensing lubricants and the greatest part of the program is accomplished.

Preventive maintenance is another factor in any lubrication program. To prevent lubrication failures is as important as repairing a failure. By this we mean much time can be spent and wasted repairing failures that otherwise could have been prevented. This, of course, costs dollars which could have been used more profitably. It is not implied that we can expect one hundred percent elimination of all failures, but they can be reduced substantially. For example, in our plants that have instituted preventive maintenance programs, our maintenance staff indicates that fire fighting jobs have been curtailed approximately 40 percent.

PROPER SCHEDULING PROGRAMS PAY OFF

Much of this success is the result of scheduling programs begun by the Maintenance Department Central Staff. Such scheduling involved the review of all equipment for lubrication requirements. This information is placed on lubrication record cards and aids in working out scheduling routes and frequencies of application.

To aid in this program, a standard identification system has been developed. This system employs decals which indicate the frequency of lubrication as well as the type of lubricant required. In this way, the oiler has the correct lubricant for each machine. These decals may also be placed on the dispensing equipment to insure against the possibility of the wrong lubricant being used. The use of such a system insures that an adequate amount of lubricant is delivered to the equipment when it is needed.

In setting up a system of scheduling lubricant applications, it should be remembered that any system must be simple and flexible. It should be simple, in that maintenance departments should not be overburdened with an excess of personnel handling records. Records should be compact and easily accessible. Oilers scheduling cards should be arranged for day-to-day requirements. It should be flexible in that changes in programming can be made easily without a complete rework of records.

We have recently completed some small scale tests making use of automatic business machines. By coding the frequency, the proper cards are released by the push of a button for each shift operation. While this test was only conducted on one phase of the over-all program, namely, oil consumption records, it was hoped the test would reveal cost data for such operations. These costs could then be projected for an estimate of total costs for preventive maintenance. In this way, anticipated savings could be compared with costs to indicate if the program was worthwhile.

One of the most significant findings, which had been suspected, is the surprising number of failures that were not due to lubrication. While maintaining a preventive maintenance program, it was found that a higher percentage of machine breakdowns were due to electrical, hydraulic, or mechanical failures. It is generally felt that, because a change in lubricant requires the least amount of work, this should be the first move in the solution of any failure. Many times such a hasty analysis of the problem results in more loss of time. The expense of replacing the same failed part after the lubricant has been changed can be prohibitive.

CENTRALIZED LUBRICATION SYSTEMS

It was mentioned briefly that we have a training course devoted exclusively to the theory, application, and maintenance of automatic lubrication systems. We have found that this course has helped immensely in acquainting those responsible for plant lubrication with the various systems manufactured.

In our plants we use centralized systems manufactured by virtually all companies marketing such systems. This was not the result of any standard practice, but rather the choice of several of our industrial equipment suppliers. Generally, we have found their choice practical. As we began to use more and more of these systems, we developed standards requiring certain characteristics for lubrication systems. These standards required manufacturers to make provisions for safety to personnel, warning signals when blockage occurred at any point and provisions for accessibility to all machine components for our maintenance personnel.

The increased use of such systems was a combination of a desire for progress and a desire to save dollars, for such methods of dispensing lubricants have proven very economical and efficient. This performance alone was the largest single factor which influenced management as to the necessity of such equipment.

Mention should be made of the significance of centralized lubrication systems in the lubricating of machine tools specifically designed for the automatic processing of parts. This process, known as automation, requires that each tool perform its operation automatically and without failure. Breakdown in a unit of an automated machine line results in a shutdown of several operations within the automated structure. In contrast, a non-automated individual machine breakdown affects only one operation until repair is made. Therefore, centralized lubrication exhibits its greatest value when correctly adapted to automated machine lines.

ESTABLISHING LUBRICANT SPECIFICATIONS

Previously, it was indicated that material specifications performed a vital part in our program. We would like to enlarge upon this subject for the benefit of those not familiar with this policy. This discussion, of necessity, is from the viewpoint of a consumer.

It is recognized that brand names are known standards of the manufacturer's quality and integrity. We, as both a consumer and a manufacturer, value the name which represents the product. However, in the manufacturing of high volumes of products, it is necessary to deviate from the buying practice of the individual consumer. This must be done in order to keep material costs to a minimum. To help us take full advantage of our purchasing power, we have incorporated specifications into our manufacturing operations. This has been done not only to sharpen our competitive buying, but also to give us a language common to all our operations.

Company material specifications also tell us the particular characteristics required of materials. In our manufacturing operations, many production and design factors necessitate the transfer of personnel between

M-6C2-A Hydraulic & Lubricating Oil Inhibited (150)
M-6C2-B Hydraulic & Lubricating Oil Inhibited (215)
M-6C2-C Hydraulic & Lubricating Oil Inhibited (300)
M-511-C Hydraulic & Lubricating Oil Inhibited (600)

1. SCOPE: The material desired under this specification is a refined mineral oil compounded with suitable additives to improve its resistance to oxidation and its rust preventive characteristics.
2. APPLICATIONS: This material is generally used as a special purpose hydraulic oil in tight hydraulic systems and as a special purpose lubricant in recirculating systems operating in a temperature range over 130° F with minimum water contamination and where gumming and oxidation must be kept to a minimum. It should not be used in systems equipped with diatomaceous earth type filters.
3. REQUIREMENTS:

		M-6C2-A	M-6C2-B	M-6C2-C	M-511-C
3.1	VISCOSITY				
	(ASTM D-88) sus @ 100° F	145-160	200-230	285-325	575-625
	(ASTM D-445) csk @ 100° F	30.9-34.3	43.2-49.7	60.3-70.3	124.5-135.3
3.2	VISCOSITY INDEX:				
	(ASTM D-567)	85 min	85 min	85 min	85 min
3.3	FLASH POINT:				
	(ASTM D-92)	357° F min	375° F min	375° F min	400° F min
3.4	FIRE POINT:				
	(ASTM D-92)	425° F min	425° F min	425° F min	450° F min
3.5	POUR POINT:				
	(ASTM D-97)	10° F max	10° F max	10° F max	15° F max
3.6	COPPER STRIP CORROSION: (212° F)				
	(ASTM D-130)	all pass			
3.7	RUST PREVENTIVE CHARACTERISTICS:				
	(ASTM D-665)	Distilled water, 24 hours — no rust Synthetic Sea water, 24 hours — no rust			
3.8	OXIDATION STABILITY:				
	(ASTM D-943)	Minimum hours before oil reaches acid numbers of 2.0, 750			
3.9	RESISTANCE TO FOAMING:				
	(ASTM D-892)	No foam			

3.10 CONTAMINATION: The material shall be free from acid, alkali, moisture, tarry or suspended matter, thickeners, or other foreign matter. It must also be free from objectionable odors.

3.11 GALLONAGE CALCULATION: Gallonage of incoming shipments shall be calculated in accordance with the methods outlined in Chart 1, Section MC99.

3.13 SUPPLIER'S RESPONSIBILITY: All material supplied to this specification must be equivalent in all characteristics to the material upon which approval was granted.

4. APPROVAL OF NEW SUPPLIERS: Materials submitted by new suppliers to this specification must have laboratory and shop trial approval.

Recommendations for the addition of new suppliers to the approved list, along with approved laboratory and shop trial reports, must be made in writing to Manufacturing Research Department by Divisional Quality Control Laboratory or Chemical Engineering Department.

Approval may be granted by Manufacturing Research, Manufacturing Staff, for divisional use or for company wide use on the basis of these reports.

Figure 1. Typical additive oil specification

Applications	Pinion Speeds rpm	Ford Material Spec.	Name	Approx. Vis. ssu @ 100° F	Remarks
Special purpose lubrication for recirculating systems, temperatures over 130 F. and water contamination.	5,000- 8,000	M-6C2-A	Hyd. & Lub. Oil, Inhibited (150)	150	Inhibited oils are more expensive than the M-8C straight mineral oils and should not be used in "once through applications" or where systems leak.
	3,000- 5,000	M-6C2-B	Hyd. & Lub. Oil, Inhibited (215)	215	
Typical applications: Machine tools, dynamos, electric motors and hydraulic pumps.	2,000- 3,000	M-6C2-C	Hyd. & Lub. Oil, Inhibited (300)	300	
	1,000- 2,000	M-511-C	Hyd. & Lub. Oil, Inhibited (600)	600	
Hydraulic oils to be selected by viscosity from pump manufacturers recommendations.	Under 1,000	M-511-D	Hyd. & Lub. Oil, Inhibited (2500)	2,500	These oils should not be used on systems equipped with "Fuller's Earth Type" filters; such filters remove certain additives.
Special purpose lubrication for high speed spindle bearings, temperatures over 130 F. and water contamination.	12,000-15,000	M-4691-K	Spindle Bearing Oil (33)	33	
	10,000-12,000	M-4691-A	Spindle Bearing Oil (65)	65	
Typical applications: Grinders.	8,000-10,000	M-4691-C	Spindle Bearing Oil (110)	110	These oils have an oxidation stability requirement (750 hours).
Pneumatic tools and air lines on presses.	—	M-2C5-A	Air Line Oil (190)	190	Water emulsifying characteristics.
Oils to be selected by viscosity from pneumatic tool manufacturers recommendations.	—	M-2C5-C	Air Line Oil (500)	500	
Air Compressors.	—	M-519	Air Compressor Oil		High detergent and soft carbon forming characteristics.
Turbines	—	M-525	Turbine Oil (150)	150	These oils differ from the M-6C2 oils only in the higher oxidation stability requirement (1500 hours).
Turbine oils to be selected by viscosity from turbine manufacturers recommendations	—	M-511-B	Turbine Oil (300)	300	
		M-2C17	Turbine Oil Marine	425	

Figure 2. Typical reference chart for the selection of noncorrosion and/or oxidation inhibited oils.

plants. We cannot tolerate a similar transfer of materials. Without specifications this would result in duplication and confusion, as brand names do not, in themselves, signify specific material characteristics.

There are, however, precautions to be taken in specification buying. The purchasing department buys material from the source submitting the lowest bid. This policy requires someone to specify quality. This may be done by publishing an approved listing of products meeting the minimum requirements. With this aid, buyers are able to guide their purchases so that only materials meeting our requirements are received.

The specific task of writing lubricant specifications has become more complex in recent years. This is due largely to the increasing number of lubricant additives available for imparting similar characteristics; for example, oxidation inhibitors or corrosion inhibitors. To offset this difficulty, performance requirements are written into our specifications. These requirements vary with the type of lubricant and the application. Despite the work involved in maintaining specifications, we have found it is a necessary prerequisite in any standardization program.

Early in our standardization of plant lubricants, we recognized two possible methods for effecting this program. One was a survey of all the equipment in our plants requiring lubrication. The other was a study of the lubricants being used, their volumes and review of the equipment manufacturer's recommendations.

The first method would involve contacting the plant engineer's office in each plant and requesting a list of all equipment in their plant. In some instances, this list was available in a usable form, but in many instances these lists would have to be compiled. The next step would be to obtain from the equipment manufacturer manuals containing data on bearings, gears, and other components in each model or series. The compilation of this data would be time consuming and laborious. It would also involve and place extra work on too many of our company operations.

The second method would involve requesting the lubricant purchase records from our non-productive materials buyer and periodic meetings with our maintenance personnel. Taking a calculated risk, we decided on the latter method for beginning our standardization program. This risk was based on the supposition that a greater percentage of the lubricants in use were recommended by the equipment builders and were adequate. In line with this thinking, we developed the following steps for the standardization and simplification of plant lubricants.

1. Classify and review all plant lubricants.
2. Review company oil purchase records.
3. Study equipment manufacturer's recommendations.
4. Develop a selection chart for these lubricants.
5. Conduct plant trials on all lubricants suspected of being misapplied.

The plant lubricants were classified as straight mineral oils, additive or compounded oils, greases, and special or intermediate compounds. To avoid any misunderstanding as to what is meant by these classifications, here are the definitions we used for the various categories.

Included in straight mineral oils were those oils derived from petroleum crude through a definite refining process, and to which no additive is incorporated, designed to impart and/or improve various physical or chemical characteristics.

Additives or compounded-type oils were defined as specially selected base mineral oils capable of accepting additives which improved or gave a characteristic or characteristics to the oil, which could not be attributed to the oil itself. A typical additive oil specification is shown in Figure 1. Additives might be used for one or more of the following reasons:

1. Improve film strength.
2. Impart oiliness.
3. Retard oxidation.
4. Inhibit corrosion.
5. Reduce foaming.
6. Impart tackiness.

Greases are a solid or semi-solid combination of mineral oils and soaps or a mixture of soaps, with or without filler, suitable for certain lubricant applications. Special or intermediate compounds, because of their make-up, cannot be classified as an oil or a grease. They are usually an asphalt residual containing fatty constituents or other materials.

The oils in each classification were grouped by viscosity, the greases by consistency or penetration, and the compounds by saybolt-fural viscosity or penetration, whichever was applicable. This listing revealed that several lubricants being used under different specifications were equivalent.

Next, we sent to our purchasing department a list of all lubricant specifications shown in our index. We asked them to supply a monthly consumption figure and the approximate cost of each lubricant. A review of their report indicated a few lubricants no longer purchased. We eliminated these specifications as soon as possible. With the information submitted by purchasing, we also made obsolete specifications of lubricants which had low consumption and were replaced with an equivalent lubricant of higher consumption. If the low consumption lubricant had a lower price, efforts were made to approve that lubricant to the specification retained. In every case where a specification was eliminated, the suppliers were given an opportunity to have their products approved under the equivalent specification to be retained. If, however, the lubricants did not meet our minimum quality requirements, they were not given further consideration.

Our next step involved a study of equipment builders' recommendations. Typical groups studied were the American Gear Manufacturers Association, The American Society of Refrigeration Engineers, the various pump manufacturers, and others. These recommendations, in addition to our needs dictated by experience, aided us in reducing the number of viscosity grades required.

Next, we reviewed all low consumption lubricants not previously considered and investigated their specific application. Where it was possible we recommended changes to more generally used lubricants of a slightly different viscosity ($\pm$50 saybolt seconds at 100°F.) This rule obviously was not applied to lubricants below 300 seconds saybolt. In the viscosity range of 60 to 300 seconds, we substituted oils differing in viscosity

(Continued on p. 120)

Mill and Laboratory Evaluation of Oils for Rolling of Copper Alloys

F. L. Reynolds, Bridgeport Brass Co., Bridgeport, Conn.

The general problem of lubricant evaluation in metal processing applications has been very difficult because of the number and complexity of conditions. Differences in metal alloys, tooling and equipment result in many rapidly changing variables.

A mill testing operation was established to investigate three major considerations of roll oils:

1. **Lubrication of strip during rolling.**
2. **Staining tendency during anneal.**
3. **Separation in recovery operation.**

Lubrication tests were conducted on the basis of temperature measurement of strip before and after rolling, utilizing a contact thermocouple. These were compared with the calculated temperatures of deformation. Stain testing was done by examination of annealed material after pickling. Demulsibility was checked by sediment and water tests after recovery of the oil.

To relieve the mill testing program, laboratory tests were devised and correlation attempted. Lubrication testing was done on a tester equipped with a steel roll running against a brass block; stain tests were made using tight, oil-soaked coils of brass strip in an electric furnace; demulsibility was checked through a modified ASTM steam emulsion test. Fair correlation, sufficient for preliminary testing and general prediction of mill results, has been obtained.

INTRODUCTION

The general problem of determining the best possible lubricant to be used in metalworking applications usually has been hampered by the difficulty in obtaining true, concise data which are directly applicable to commercial operating conditions.

The most important and most difficult part of testing on production equipment is identifying and controlling all variables so that a useful and informative testing operation can be set up. The variables include a wide range of tool surfaces, metal properties (including surface and temper), methods of applying lubricant, working speeds, and temperatures. Meaningful testing demands rigid control over every item.

Because of these difficulties, the practice in the past has been, usually, simply to obtain large samples of a particular lubricant to be tested, try it in the particular operation under consideration, and see what happens. Unless there is a spectacular difference between the tested oil and the reference lubricant, it bcomes very difficult to establish any concrete conclusions. As a general rule, there is usually only a small difference and even though the difference may be quite important, the better lubricant does not get the proper consideration.

Presented before the 12th Annual Meeting of the ASLE, April 17, 1957 at Detroit, Mich.

CONSIDERATIONS IN ROLLING

In the particular case under consideration, that of rolling flat sheet and strip, three main considerations must be made: lubricant effectiveness or load carrying ability, resistance to staining during anneal, and demulsibility in reclaiming operations.

The problem of load measurement is pertinent to its action at the roll stand only; the stand used for these tests is shown in Figures 1a and 1b. Figure 1a shows the

Figure 1a. Entrance side of roll stand used for tests.

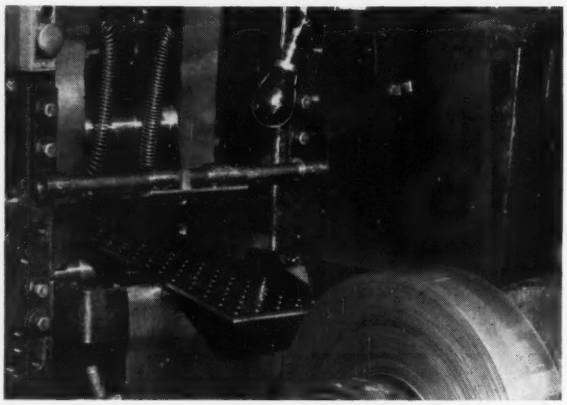

Figure 1b. Exit side of roll stand for tests.

rack used for uncoiling the strip, the wipers through which the strip passes, the guide box with its wooden wedge for maintaining even tension, the oil sprays (directly above the guide box), the water sprays (flexible hose down from horizontal header above box), the rolls, and the roll adjustment screws on top of the stand. Figure 1b illustrates the exit side of the rolls and the recoiling apparatus.

The temperature of the rolled sheet is taken to be a measure of lubricant effectiveness in the rolling operation. In addition to dependence on the lubricant, it is also dependent upon roll setting (screw down pressure), roll finish or surface condition, cooling water volume and temperature, torque developed by coiling equipment, roll speed, alloy and alloy condition. It is important to measure sheet temperature, roll temperature and the power developed to the rolls.

The problem of staining during the annealing cycle is important because of the necessity of maintaining clean metal and the difficulty arising in subsequent pickling.

Figures 2a and 2b illustrate the types of stains generally obtained with the lower end of each strip having been cleaned to show the original base metal. While difficult to show in a black and white photograph, the stains do have varying characteristics. Figure 2a (left) is a severe example of thick varnish deposits which cover the metal completely, varying slightly in thickness of film and color from the solid shiny black near center to the thinner, color refractive film just below the top. Figure 2a (right) is similar, but more evenly distributed and generally thinner. Both films are capable of almost complete insulation of the metal from pickling acid and generally necessitate reannealing to burn off the stain, thereby also interfering with metallurgical factors. Figure 2b (left) illustrates a stain which consists primarily of a

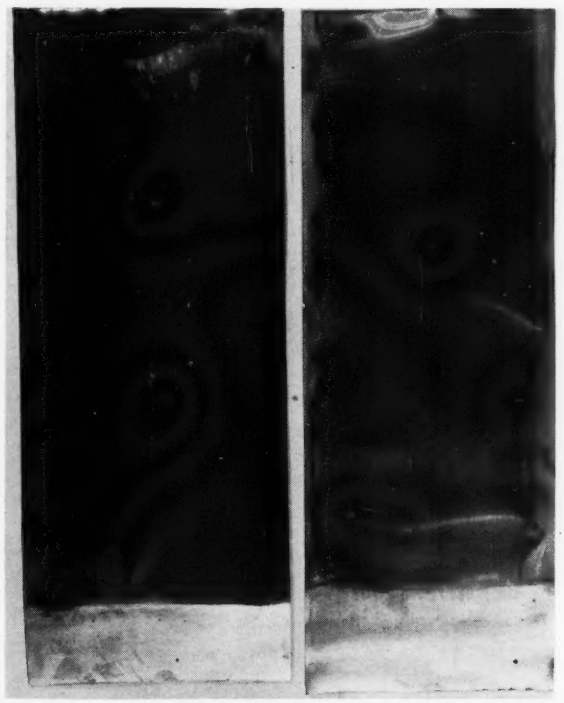

Figure 2a. Types of oil stains obtained in mill production.

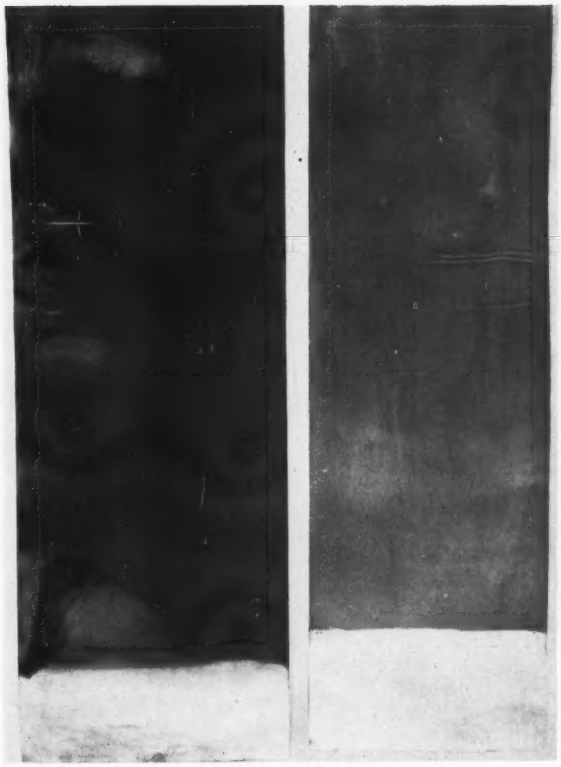

Figure 2b. Types of oil stains obtained in mill production.

sooty coating of sufficient thickness to severely discolor the metal, but is porous enough to permit acid penetration and does not present too serious a problem. Figure 2b (right) is similar, but to a considerably smaller degree, and is a general situation causing no trouble.

The problem of demulsibility depends to a great extent on the type of reclaiming system vs the volume of water and oil which have to be separated. It also depends on mill conditions, insofar as the amount of dirt from the air and floors and attritus from the rolling operation is concerned, because of the stabilizing effect of small particles on any emulsions formed and on the demulsibility tendencies of the oil.

DEVELOPMENT OF MILL TESTS

In the development of actual tests for mill use, it became necessary to consider each phase in the light of practicality and economy. The procedure adopted had to be applicable to routine usage throughout the mill if necessary.

Measurement of Load Carrying Ability

The basis of the test became a temperature measurement problem, complicated by the necessity of heating or cooling the rolls with steam or water to maintain shape and flatness of the rolled metal.

The temperature of the metal did not appear to be affected by the roll temperature as long as the variation of the roll temperature during a given pass was not more than three or four degrees. By allowing the rolls to dissipate surface heat into their own core, rapid fluctuations of surface temperature were not evident and the effect on

rolled metal was reduced to a minimum while still maintaining hot rolls. The lower temperature (115° F) compared favorably with standard practice while the end point temperature (highest was 168° F) served to make the test more severe. Through the use of steam or water the temperature over the length of the rolls could be adjusted to a given value before the test, and the temperature of the rolls was then allowed to rise normally.

The use of load cells in the hold-down screws was ruled out, as were torque measurements. However, each roll stand was already equipped with an ammeter registering current in the main driving motor and a tachometer calibrated in feet per minute of the roll surface.

The surface roughness of the rolls was checked with a commercial surface analyzer and found to be constant over a given small run of perhaps three or four coils of strip. Emery cloth used for manual roll polishing between runs was found to essentially reproduce the same surface each time it was used.

The reproducibility of roll setting was found to be most easily done by the use of sample strips usually taken directly from the metal being used in the test.

After a number of trials, a test was devised along the following lines: the roll temperature was stabilized at 115° F at the start of each run with no water flowing over the rolls during the test. The test was then begun and temperatures of the strip going into the mill, the strip coming off the mill, and the rolls were measured. A two-prong contact thermocouple designed for the purpose was used in conjunction with a potentiometer calibrated from 50 to 250° F. Readings were taken of the power (watts) and speed of the mill after full running speed had been attained. It was found that, during the several seconds before start-up and after stopping, the temperature of the metal in the coil did not change appreciably and was not significant in effect.

The type of result obtained is illustrated by the following example: phosphor bronze strip 8 in. by 0.050 in. was rolled in five passes from a fully annealed condition to a 50 percent reduction. The metal was divided into four, three-coil lots, and each lot was fully rolled using an oil under test.

All three bars of each lot were rolled successfully at each reduction without control over roll temperature after the initial adjustment to 115° F before the start of the entire run. Temperatures of the roll were recorded after each pass and the metal temperature before and after each pass. Power and speed readings were taken after the mill had stabilized.

Using the basic formula of metal deformation,

$$W_k = KQ \ln G_0/G_1 \qquad [1]$$

where

W_k = work of deformation
Q = volume of metal flow
K = flow stress
G_o = original metal thickness
G_1 = final metal thickness

and equating this to its equivalent heat value,

$$W_k = Q\rho c_p \Delta t \qquad [2]$$

where

ρ = density
c_p = specific heat
Δt = temperature rise during deformation

the work equation becomes,

$$\Delta t = \frac{K \ln G_0/G_1}{C_p\rho} \qquad [3]$$

For the purpose involved here, the respective values became

K = 1.15 (.05% Yield Strength)
ρ = 0.322 lb/in.^{3}
c_p = 1.62 Btu/lb—°F

and the final equation, including conversion factors, as used in the test, for theoretical temperature of deformation

$$\Delta t = 0.000237 \ (0.5\% \text{ Y.S.}) \ \ln G_o/G_1 \qquad [4]$$

Subtraction of this temperature change from the apparent temperature changes as actually determined during the test gave temperature figures proportional to friction losses, Δt_f.

It was found that the inclusion of roll temperature in the calculations did not effectively change the results, apparently due to a relatively low actual heat flow between strip and rolls. Possibly, a more accurate determination could have been obtained by the addition of roll temperature change. However, inasmuch as measurements of heat transfer through the rolls, the metal roll interface, and their bearings would have been very difficult, it was decided not to incorporate this data directly into the equations but to retain it as a control measure. These figures of Δt_f when plotted against the percent reduction gave curves as in Figure 3. Also included are the approximate actual metal temperatures obtained at those reductions.

It should be noted that there are several cross-over points in these curves, the precise reasoning for which is unknown. However, the most important factor to consider would appear to be temperature vs additive reaction. Inasmuch as the metal was rolled consecutively, i.e., the second pass immediately after the first, etc., according to standard practice, a cumulative temperature rise was encountered. From the known composition of these oils, is is readily possible that the wetting or surface reaction between base metal and oil additive would depend on the temperature and, in fact, would reach different levels in different oils at various temperatures.

Also to be considered are the roll pressure and metal thickness, particularly during the last passes, as evident in the curves. Although the work hardened material re-

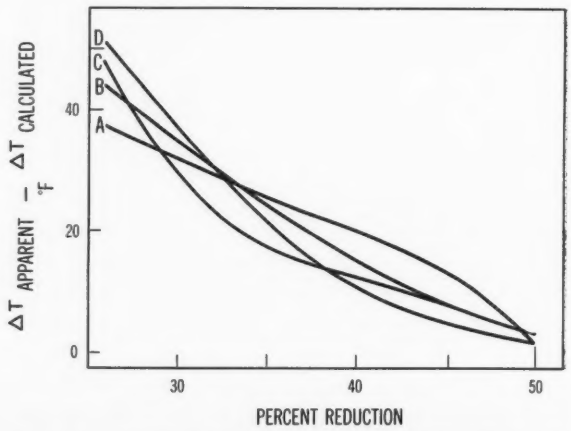

Figure 3. Apparent temperature change minus calculated temperature change due to deformation vs % reduction.

quired higher pressures in the last passes, in the neighborhood of 75,000 to 80,000 psi, the thickness ratio of incoming to outgoing metal approaches one, reducing the frictional area. Therefore, while the temperature of deformation in these instances was of the order of 12 to 15° F, the temperature rise due to friction was only a matter of one or two degrees.

In an attempt to prove or disprove these curves, a plot of the current-speed ratio vs the percent reduction was made, as in Figure 4, showing that the higher power demands coincided with the higher frictional losses as measured by temperature.

Measurement of Stain Resistance

The evolution of a mill test for staining tendency of an oil was very much complicated by the wide number of possible conditions which could exist. Variations encountered included the alloy, tightness of coil during anneal, temperature of anneal, and furnace atmosphere during anneal.

The alloy involved did not affect the actual oil stain, rather, the type of oxide formed, i.e., loose or tenacious, through a particular alloy — atmosphere — temperature combination, determined the ease of removal of any stains formed by the oil.

The actual formation of the stain was primarily determined by the oil temperature — atmosphere combination. In general, the higher annealing temperatures in the range of 550 to 650° C produced less stain, whereas at low temperatures of 400 to 500° C troublesome stains were produced with many oils, particularly with oxidizing or so-called neutral atmospheres. Less trouble was encountered with reducing atmospheres apparently because the distillation of the oil film progress further before carbonization, polymerization, or oxidation of the oil reached a high level.

The only manner in which a mill test could be conducted was to run the entire mill on one oil and examine almost every bar and coil of metal as it was pickled for a period of at least a week. However, because it was primarily a case of "clean or not clean" after pickling, it became a simple matter for the foreman of each shift at the pickling units to record any facts concerning metal

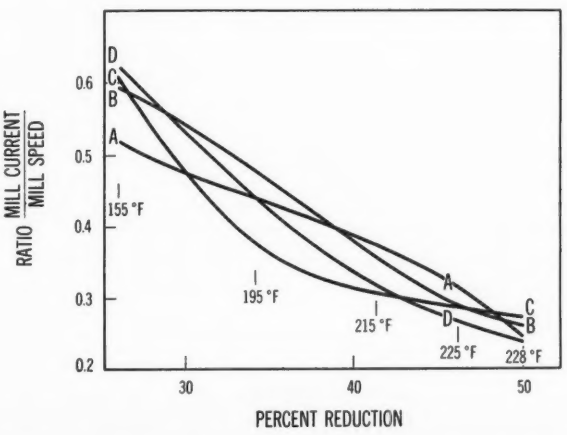

Figure 4. Ratio mill current to mill speed vs % reduction.

which had been badly stained and was not satisfactory even after pickling. It was found that this method worked quite satisfactorily and resulted in a good comparison of the oils with regard to their staining tendency.

Measurement of Demulsibility

The process for determining the necessary demulsibility rating was dependent primarily on the amount of time and effort required to separate a given quantity of oil, water, and attritus mixture and comparisons of resulting oils. The reclaiming process, which involved heating, settling, reheating and centrifuging, being more or less standardized, was operated normally during testing periods and records of sediment and water content of the finished oil showed whether or not the oil could be suitably separated.

DEVELOPMENT OF LABORATORY TESTS

In order to adequately investigate larger numbers of lubricants without interfering with mill production, it became necessary to attempt the establishment of laboratory tests which would correlate fairly well with the results obtained in mill testing operations.

Measurement of Lubricant Effectiveness

A test for lubricant effectiveness was devised based on the rate of wear of a brass rider at low loads, rather than scoring at high loads. A testing unit was constructed, as shown in Figures 5a and 5b, so designed that a steel roll revolving with its lower edge in an oil bath was being acted upon by a brass block carried in a loading arm above the roll. Temperature of the oil bath was controlled to plus or minus 1° F by means of electrical heating or water cooling.

Electrical Conductance Across the Lubricant Film

The brass block carried by the loading arm was electrically insulated from the loading arm so that an electrical circuit could be made from the brass block through the rotating roll by means of a lead to the block and the carbon brush on the roll shaft. The junction between the block and roll was then placed in series with a dry cell, a milliameter, and a potentiometer control. When arranged in this manner, with the potentiometer adjusted to give a dead short current during rotation of the roll shaft of 10 milliamperes, the milliameter then became a reliable indicator of the type of condition existing between roll and block. When readings of 9 to 10 milliamperes were obtained, a very definite and severe wear took place, which, if allowed to continue for more than two or three minutes, rapidly developed into deep scoring. Readings in the range of 6 to 7 milliamperes generally indicated an average condition which was determined to be a safe operating zone, while readings of 5 milliamperes or below were obtained only under unusually light load or low temperature situations.

Runs were made on individual oil samples for periods of one-half hour using a beam load of approximately sixty pounds. Roll speed was approximately 458 feet per minute. Temperatures were increased approximately 10° F for each successive run on a particular sample.

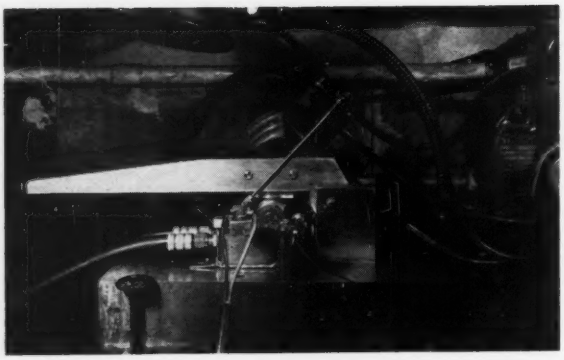

Figure 5a. Lubricants testing machine (Exposed.)

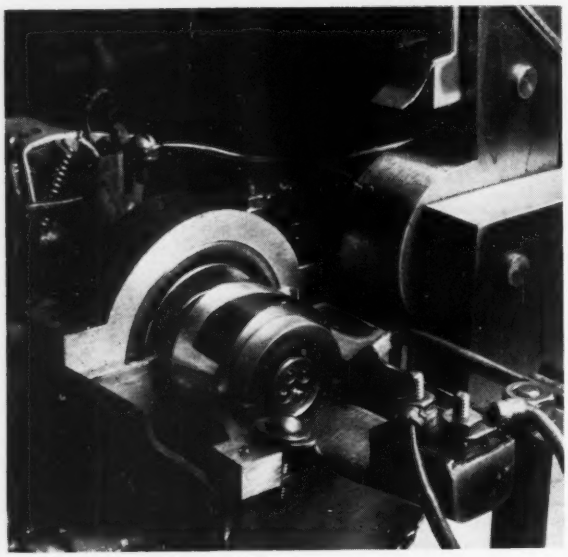

Figure 5b. Lubricants testing machine (Loaded.)

the area worn into the brass block being measured after each run. The areas were measured under 8X magnification, using a planimeter. After repeated testing, it appeared that the average oil ran 6,000 to 7,000 psi, and so-called "good oils" carried 8,000 to 10,000 psi in this procedure.

Curves for the same four oils as shown previously in the mill tests are plotted in Figure 6. It should be noted that the temperatures obtained in these tests are lower than in the mill tests because of limitations in the laboratory testing apparatus. However, Oils A and B show an entirely different characteristic from Oils C and D, of the same type as in the curves obtained in mill tests. Oils A and B, while showing as better lubricants insofar as wear area is concerned to the extent of the laboratory tests, show a rapid increase in wear rate as temperature is increased, whereas Oils C and D, showing a higher initial wear, indicate a decreased wear rate with increased temperature. This type of crossover is of the same nature as shown in Figure 3 for the mill tests.

From these results it is apparent that the rate of increase in wear area must be considered in addition to the actual areas. Whether or not the curves obtained would actually duplicate the mill test curves is not known as laboratory tests have not been run to the highest range of temperatures. However, results have indicated that sufficient information can be obtained through consideration of both wear area and rate to reliably predict performance of rolling oils.

Measurement of Stain Resistance

In the development of a laboratory stain test, an attempt was made to follow as closely as possible the actual conditions which generally exist during mill annealing operations. A general temperature range of 400 to 600° C was selected and various shapes and sizes of metal strip were covered with oil and heated at these temperatures for given times in an electric furnace. Also introduced were several different atmospheres to obtain either oxidizing, reducing, or essentially neutral atmospheres.

For the most part, flat plates soaked with oil and sandwiched together did not give reproducible results except at very short time intervals, in the neighborhood of five to ten minutes at temperature. It was generally found that the etch effects, which were more or less present depending on the atmosphere, extended too far to the center of small plates to permit accurate determinations.

The best method appeared to be that in which a small coil of metal was used, and subsequently a coil of 85-15 copper zinc, 0.015 in. thickness, 2 in. width, and 5 ft long, was standardized on. This strip was coiled loosely over a 3/8 in. arbor, immersed in the oil to be tested, and then coiled tightly and bound before removing the arbor. Further tests were run between 400 and 600° C using either air or natural gas as an atmosphere. Insofar as varnish types of deposit were concerned, there was a noticeable difference with temperature and with atmosphere.

The trend seemed to be less varnishing at the higher temperatures than at the lower temperatures, with practically none above 550° C in a natural atmosphere. With an oxidizing atmosphere, considerably more was noted at 400° C than with the neutral atmosphere, with appromimately the same result above 550° C. The final test involved the use of the coil as described above, placed in a furnace at 500° C for a period of forty-five minutes, using

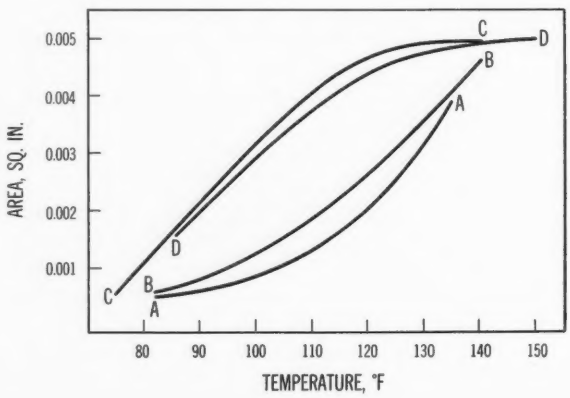

Figure 6. Wear area, lubricants testing machine vs temperature.

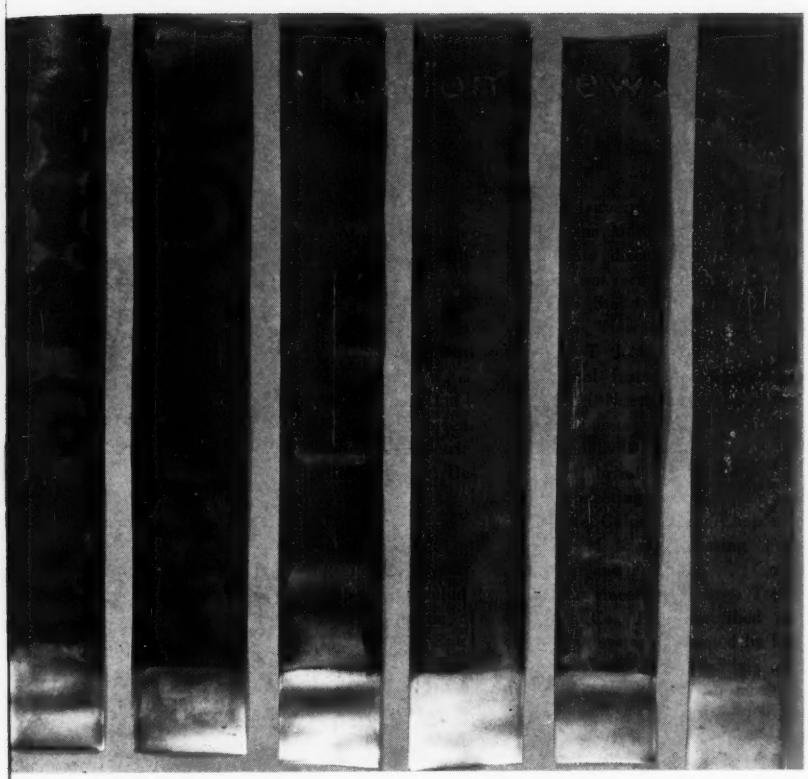

Figure 7. Samples of stained metal, laboratory stain test,

the neutral gas as an atmosphere in the furnace. These coils were examined after cooling in air without being cleaned in any way. The results which were obtained showed generally more stain than was the average situation in the mill production. However, of a group of oils, each seemed to retain its relationship to the others in both cases.

The most severe type of stain encountered, from the cleaning standpoint, has been the varnish type of stain, whereby a thin film of partially carbonized and polymerized oil remained on the metal, preventing wetting of the metal by the acid pickle solutions. Variations of this type of stain vary from a light brown almost transparent film capable of refracting light to the hard black coating similar to black enamels.

Insofar as varicolored stains were concerned, it appeared that the majority, other than those due to thin transparent films of varnish, were caused by additives of various types which formed colored reaction products with either the copper or zinc. It was found that in the majority of cases where the light colored stains were obtained in the laboratory tests, very little difficulty was encountered in the mill and they were easily pickled off.

Illustrated in Figure 7 are six sections of strip from stain tests. Numbering from right to left, Samples 1, 2, and 3 show increasing degrees of coloration due to thin films of varnish, Sample 4 shows a red blush frequently obtained due to slight additive reactivity and dezincification, and Sample 5 is a cleaner version of the same showing very little varnish or stain. Sample 6 is severely etched and stained, due primarily to strong additive reaction.

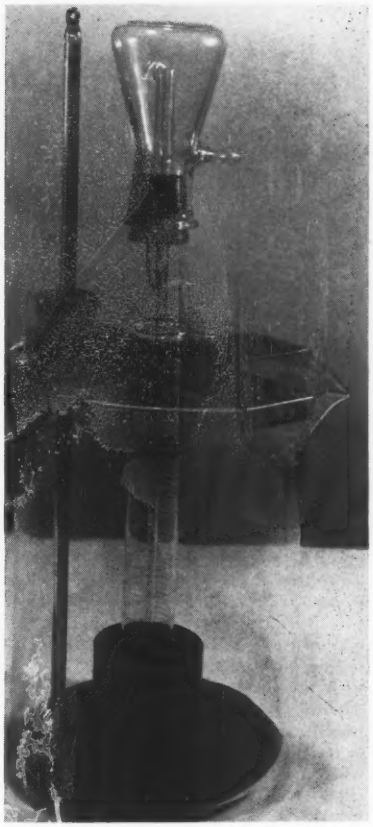

Figure 8. Glassware used for steam emulsion test.

Measurement of Demulsibility

In the development of a test for demulsibility, an attempt was made to use the ASTM Test D157-36, for steam emulsion, which has since been discontinued as standard by the ASTM.

In the course of experiments with this test, it was found that only fair reliability could be obtained at SE numbers between 150 and 250, while above this the results were very erratic. In an attempt to stabilize the procedure, the source of steam was changed from a one-liter glass flask to a standard 10 psi steam line delivering steam from the main power plant.

A 50-milliliter suction flask was arranged to provide a steam trap at the upper end of the delivery tube and means were supplied for drawing off water condensate as shown in Figure 8. While this method did present the possibility of introducing contaminated steam, the disadvantages were more than offset by the increase in controllability and constancy of the steam supply. While the temperatures of the water baths were maintained the same as the standard test, a change was made in the volume of oil used as sample. Instead of a 20 milliliter charge, 23 milliliters were used, although 20 milliliters were still separated out as before. With one operator the results in this procedure have shown a consistent repeatability with results seldom varying more than five seconds and consistently less than three percent in the range being used.

With the system of reclamation in use, it was found that in order to obtain oil acceptable under the sediment

(Continued on p. 120)

Grease Flow in Shielded Bearings

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A technique, based on dye tracers, for measuring soap and oil components of a grease in shielded bearing lubrication has been developed. Application of this technique to small, high speed bearing tests has shown that there is substantial movement of grease between the bearing proper and shields. The amount of grease movement at ambient temperatures as determined by tracing oil flow is identical with the flow determined from soap tracer determinations. This indicates that the grease is moving between the bearing and shields as an entity. Considerably more work is needed to investigate quality factors in bearing greases which are probably related to flow. The techniques developed should be extremely useful in such studies.

INTRODUCTION

The mechanism of grease lubrication of shielded bearings has been the subject of considerable speculation but the object of very little experimentation. The most widely espoused theory is that the soap portion of the grease acts as a sponge for the oil portion and releases the oil to the bearing at a sufficient rate to provide lubrication. The work reported here has been undertaken to provide quantitative information on the flow of grease as an entity and on the flow of the oil component in a shielded bearing. An understanding of what actually happens in a bearing is a necessary first step in elucidating the mechanism of lubrication.

PRESENTATION OF DATA

Procedures and Materials Used

The method used to follow grease movement in small, shielded bearings involved the use of dyed grease in the bearing and undyed grease on the shields, and quantitative determination of the extent of dilution of the dye in the grease in the bearing after a test run. The procedure consisted of running two 204 bearing assemblies (ABEC-NLGI apparatus), one containing 3 grams of dyed grease and the other 3 grams of undyed grease, for 2 hours at 10,000 rpm to distribute the grease in an equilibrium position. The bearing assemblies were then dismantled and the distribution of grease between the bearing and the shields determined by weighing. The set-ups were then reassembled and the shields with the undyed grease were used with the bearing containing the dyed grease. The new assembly was then run for two additional hours under the conditions used for the original distribution of grease. The dye content in the bearing was then determined and the movement of grease calculated from the dilution of the dye in the bearing.

The dyes used were sodium fluorescein and an oil soluble blue dye. The sodium fluorescein was added as a 5 percent solution in water and was blended into the cold grease to the extent of 100 parts per million. The oil-soluble blue dye was added as a 10 percent solution in benzene and 200 parts per million based on the grease was used. Dye contents were followed by dispersing the traced greases in chloroform, precipitating the soap with alcohol, centrifuging to obtain a clear supernatant solution, adjusting the total volume, and analyzing for the dye. The pH was adjusted to 10 minimum by addition of alcoholic KOH if necessary, since fluorescein absorption is affected by pH. The solutions were analyzed in a recording double beam spectrometer using a chloroform-alcohol solution as a blank. Peaks above the base line absorption measured the concentration. Both dyes follow the Lambert-Beer law, and are independent of each other over the concentration range studies. This means that the log of the light absorbed by each dye is directly proportional to the dye concentration. The sodium fluorescein peak absorption is at approximately 500 millimicrons and the oil blue dye at 600 and at 650 millimicrons. Details are given in the appendix.

The sodium fluorescein is strongly adsorbed on the soap and serves as a soap tracer, while the oil blue dye is in the oil phase. This was determined by pressure oil separation tests and analysis of dye content in the oil. As shown in Figure 1, the separated oil contained the oil blue dye in the same concentration as the original grease but contained no sodium fluorescein.

Two greases of the sodium complex soap type were used. Soap contents were 16 and 32 percent respectively. Penetrations were 380 mm/10 and 210 mm/10 as charged to the bearing. Both greases harden markedly from the shearing action in the bearing, and in actual use after the 2-hour run-in, the penetrations were 240 mm/10 and <150 mm/10, respectively. These data are shown in Table 1.

Presented at the 11th ASLE Annual Meeting, April 4-6, 1956, Pittsburgh, Pennsylvania.

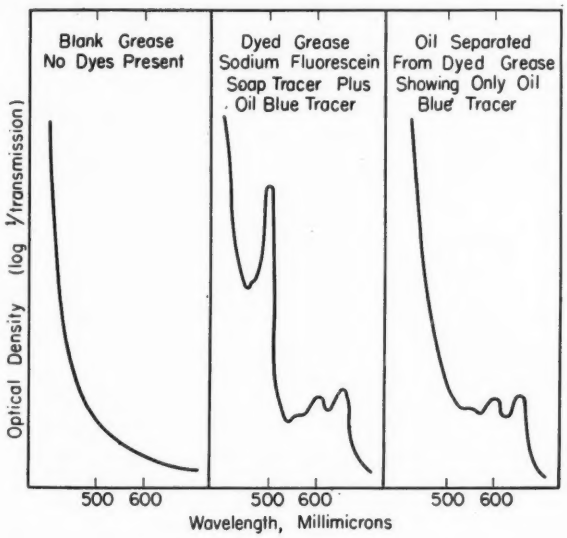

Figure 1. Grease dye studies spectrophotometer analytical curves.

TABLE 1. GREASES USED IN FLOW STUDIES

Grease	Type Soap	Penetration as Charged[1]	Penetration in Use[1]
A	Sodium Complex	380	240
B	Sodium Complex	210	<150

[1] Measured in Tenths of a Millimeter

The equipment used to measure the penetration of the greases after the run-in period is discussed in detail by the same authors in another paper[1].

Test Results

As mentioned above, the determination of grease flow in the bearing assembly was based on the quantitative analysis of the dilution of dye in the bearing grease. The relationship between dilution of dye in the bearing grease and the grease flow from the shields to the bearing is dependent upon the distribution of grease between the bearing and the shields. The greater the amount of grease in the bearing proper, the less would be the dilution of dye in the bearing for a given amount of grease movement from the shields. The first step, then, in calculating grease flow from dye dilution measurements, is to determine the distribution of the grease between the bearing and shields.

Penetration of the grease affects its distribution between the bearing and shields. The softer Grease A leaves a greater amount of grease in the bearing proper under dynamic equilibrium conditions than does Grease B. The equilibrium distribution is reached relatively quickly with these greases (after only 15-30 minutes) but to assure equilibrium the run-in periods prior to exchanging shields and bearings were extended to 2 hours. Data on distribution after the 2-hour run-in period are summarized in Table 2.

TABLE 2. GREASE DISTRIBUTION IN BEARING ASSEMBLY (3-GRAM CHARGE, 10,000 rpm, 2-Hrs.)

Grease	Pen. in Use[1]	Grease in Brgs.	Grease on Shields
A	240	1.8	1.2
B	150	1.0	2.0

[1] Measured in Tenths of a Millimeter

The dye content of the grease in the bearing showed a marked reduction after the 2-hour test run with undyed grease in the shields. In the case of the sodium fluorescein tracer, the dye content in the bearing grease dropped from the original 180 x 10^{-6} grams to 115 x 10^{-6} grams with Grease A and from 100 x 10^{-6} grams to 71 x 10^{-6} with Grease B. The percentage drop in concentration for the oil blue dye was essentially the same (within $\pm$ 10 percent of the percentage decrease in dye content). These dates in Table 3 show definitely that the soap is involved in the flow and that the grease is moving as an entity.

TABLE 3. TRACER CONTENT OF GREASE AFTER FLOW TEST

	Tracer Content*					
	Na Fluorescein			Oil Soluble Blue Dye		
Grease	Before	After[1]	% Decrease	Before	After[1]	% Decrease
A	180	115	36	360	225	38
B	100	71	29	200	135	32

[1] After 2-Hr. 10,000 rpm Test with 3-gram grease charge.
*In Millionths of a Gram

The extent of flow can be estimated from the dye content dilution by a graphical integration procedure. This calculation assumes the idealized condition that the grease on the shields is uniform in dye content and that the grease in the bearing is also uniform. Stepwise calculations of dye contents were made in which successive 10 percent increments of the grease in the bearing were transferred to the shields and an equivalent weight of grease from the shields was transferred to the bearing. Sample calculations are shown for Grease A in Table 4.

TABLE 4. DYE CONTENT IN BEARING vs. GREASE FLOW (100 rpm DYE IN BEARING GREASE

GREASE A				
Flow, g	Grease in Bearing, g	Dye in Bearing*	Grease in Shield, g	Dye in Shield*
0	1.8	180	1.2	0
0.18	1.8	180 − 18 − = 162	1.2	18
0.36	1.8	162 − 16 + 3[1] = 149	1.2	31
0.54	1.8	149 − 15 + 5 = 139	1.2	41
0.72	1.8	139 − 14 + 6 = 131	1.2	49
0.90	1.8	131 − 13 + 7 = 125	1.2	55
1.08	1.8	125 − 13 + 8 = 120	1.2	60
1.26	1.8	120 − 12 + 9 = 117	1.2	63
1.44	1.8	117 − 12 + 9 = 114	1.2	66

*In Millionths of a Gram.
[1]Dye returned to bearing from shield is calculated by multiplying dye in shield at conclusion of previous transfer by 0.18/1.2

A curve of dye content versus grease flow can then be produced from calculations of dye content after each increment of grease movement.

It can also be shown that the assumptions of complete mixing given above lead to the expression

$$F = \frac{WW'}{W + W'} \ln \frac{W'C_0}{(W + W')\,C - C_0W}$$

where C is the concentration of dye in the bearing after an amount F has been circulated to and from the shield,

W is the weight of grease in the bearing,

W' is the weight of grease in the shields, and

C_0 is the initial concentration of dye in the bearing.

A working curve for each grease may be constructed using either the graphical or formula method based on the equilibrium distribution of the grease. The flow of grease in the test run can be read from this curve, knowing the final dye content of the grease in the bearing. This is shown graphically in Figures 2 and 3. The grease flow from the shields to the bearing for Grease A as read from Figure 2 is 1.4 grams. Figure 3 indicates that 0.36 grams of Grease B flow from shield to bearing under the conditions of this test. This amount of flow in a 2-hour period is quite surprising in view of the hard penetration of these greases in use.

The flow estimated from Figures 2 and 3 is the amount of flow which would account for the dilution of dye in the bearing under the idealized situation of completely uniform bearing grease and completely uniform shield grease. This estimate of flow is the minimum amount which would account for the dilution of dye in the bearing. Non-uniformity of grease on the shield due to incomplete mixing of grease and dye which has flowed from the bearing would affect the flow estimate, since this is a deviation from the idealized situation assumed. An additional experiment was therefore run to determine the extent of deviation from the ideal. Results showed that the over-all conclusions regarding flow would not be altered significantly by the deviation from idealized conditions, as discussed below.

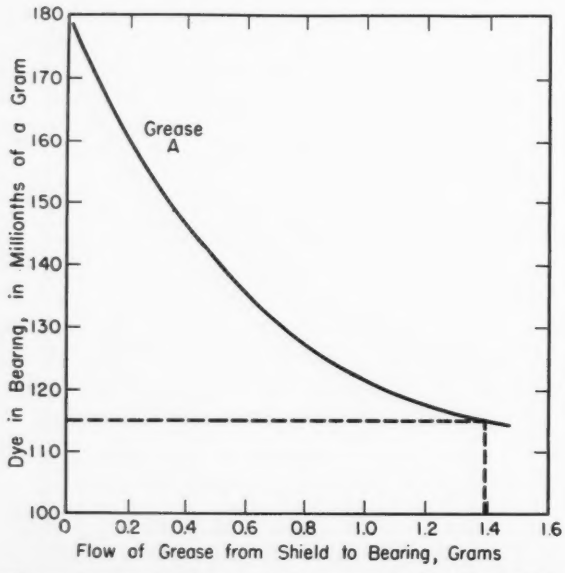

Figure 2. Grease flow vs. dye content in bearing (2 hr, 10,000 rpm, 3 g charge.)

Dye contents were determined on grease samples sectioned in small layers from the grease on the shields. These determinations showed that in the case of Grease B about 85 percent of the dye in the shield grease is located in the 0.5 grams of grease located immediately adjacent to the bearing. This indicates that only 0.5 grams of grease rather than the complete 2 grams of grease located on the shield had participated in the transfer with the bearing grease. Under this condition, the dye content of the grease in the bearing is not being diluted as rapidly for a given amount of flow because of the fact that the grease coming into the bearing from the shield has a higher dye content than it would have under the idealized condition of completely uniform shield grease. An estimate of the flow can be arrived at by recalculation based on the assumption that only the 0.5 grams of grease is involved in the mixing. This estimate would more closely approximate maximum flow. The recalculated flow vs. dye content curve based on this assumption is shown in Figure 6 from which it is seen that 0.6 grams flow of grease between shield and bearing would account for the dilution of dye noted in the bearing grease. The best estimate of grease flow then for the two greases under study is as shown in Table 5.

TABLE 5. GREASE FLOW FROM SHIELD TO BEARING (2 HRS. 10,000 rpm, 3 GRAM CHARGE)

Grease	Flow, Grams
A	1.4, Minimum
B	0.4 — 0.6

The flow of Grease A may be greater than 1.4 grams. No data are available on the uniformity of the dye concentration on the shields from which a maximum flow figure could be estimated. It is believed, however, that Grease A on the shields would be more uniform becase it is softer and should mix more easily than Grease B.

This amount of flow in 2 hours is, of course, entirely sufficient to provide good lubrication. It has been estimated that only 0.0005 grams per hour are needed to lubricate a bearing with a fluid lubricant.

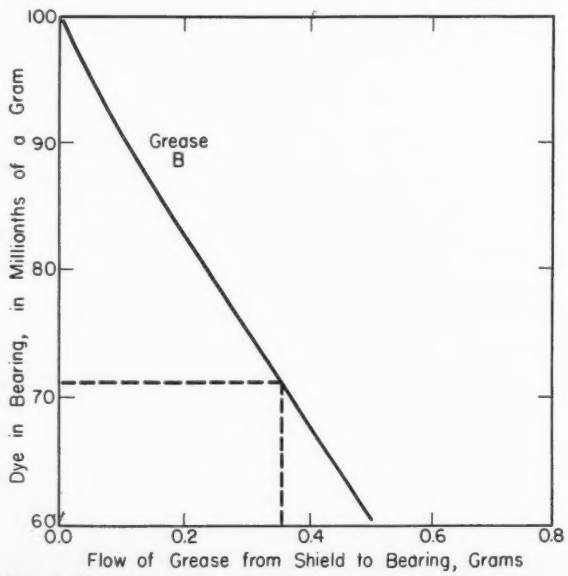

Figure 3. Grease flow vs. dye content in bearing (2 hr, 10,000 rpm, 3 g charge.)

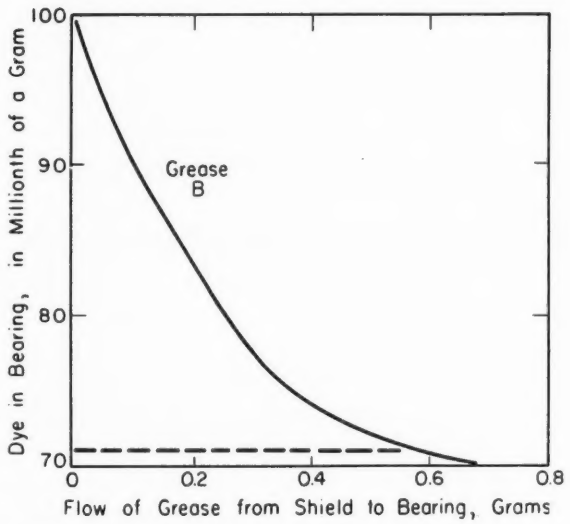

Figure 4. Grease flow assuming incomplete mixing on shields.

There are undoubtedly many quality factors in grease lubrication of shielded bearings which are related to grease flow. Temperature rise characteristics, torque requirements and even lubrication life are probably related to flow and/or grease distribution. Also, grease flow is undoubtedly affected by temperature. Work conducted to date is not sufficient to define these relationships. The technique described, however, should serve as a useful tool for future studies concerned with the mechanism of grease lubrication of bearings.

ACKNOWLEDGEMENT

The authors appreciate the work of Dr. Hugh H. Horowitz of the Esso Research and Engineering Company who derived the mixing expression given in the paper.

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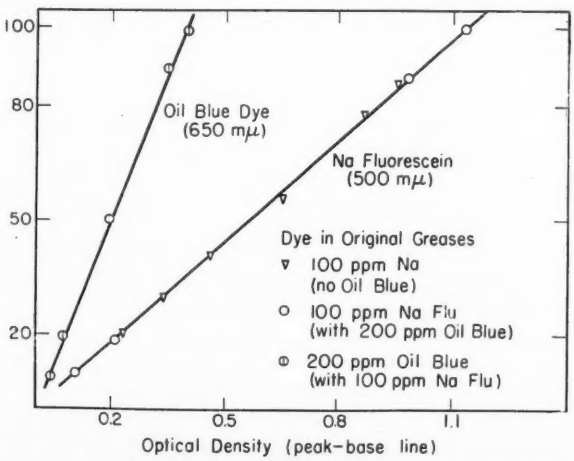

Figure 5. Optical density of extracted dye solutions from blends of dyed and undyed grease.

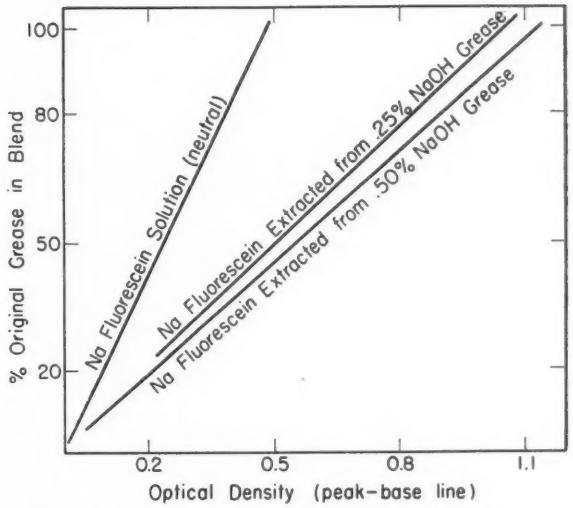

Figure 6. Optical density as affected by free alkali.

APPENDIX I. SEPARATION AND ANALYSIS OF DYES IN GREASES

A tracer grease is prepared by dispersing 0.01% sodium fluorescein (5% in water) and 0.02% oil blue dye (10% in benzene) in the blank grease with a low speed mixmaster. The fluorescein is adsorbed by the soap and the blue dye remains in the oil. The tracer grease is used to establish a working curve by making 5-10 samples of different tracer content (with blank grease diluent). These samples are treated as experimental samples; dispersed in chloroform, boiled down to 10-fold volume; transferred to centrifuge tube, equal volume of ethyl alcohol added, and centrifuged to precipitate the complex sodium soap. It may be necessary to use a higher proportion of alcohol to precipitate less dense soaps. The transparent supernatant fluid is protected from evaporation and light until analyzed. For acid or neutral greases, it is necessary to add alcoholic KOH to bring the alkalinity back to a pH of 10 minimum, in order to get maximum absorption at 502 mμ.

The test samples are run in a recording double beam visible spectrophotometer (400-700 mμ) against a blank chloroform-alcohol mixture. Peaks at 502 mμ (fluorescein) and 600 and 650mμ (blue dye) are measured above their base line. Curves of dye content vs. absorption are then drawn. The absorption data indicate that:

1. Both dyes follow the Lambert-Beer Law in this concentration range as shown in Figure 5.

2. Fluorescein and the oil blue are independent of each other as shown in Figure 1.

3. Fluorescein is affected by alkali content (Figure 6).

These working curves are used for picking off the concentration of tracer in experimental samples. A new working curve is drawn up for each traced grease.

DISCUSSION

by Dr. E. G. Jackson, General Electric Co., Lynn, Mass.

The authors have used an ingenious technique for studying the flow of grease in shielded bearings. At first presentation, their data seem to contradict the theory that grease serves as a storehouse for oil and that the oil is the active lubricant.

Looking at the dimensions of this bearing, however, reveals that it is a special system, an important but not general case.

From the author's data, I calculate that the thickness of grease if spread over the whole shield would be about two millimeters. The minimum clearance between the shield and the ball retainer is about one millimeter. Of these small dimensions, the half millimeter next to the balls where some plowing and stirring must occur, picks up 85% of the dye, leaving 1.5 millimeters relatively undisturbed. *(Continued on p. 117)*

Analysis of Equilibrium Operating Temperatures of Railroad Journal Bearings

W. M. Keller, Association of American Railroads, Technology Center, Chicago, Illinois

G. L. Pigman, Caterpillar Tractor Co., Peoria, Illinois

A graphical method of analysis is described for determining equilibrium operating temperatures of railroad journal bearings under various practical operating conditions. The method utilizes existing laboratory data describing friction characteristics of the railroad bearing, friction properties of the waste pack lubricator, and heat dissipation characteristics of the bearing assembly.

In specific examples of use of this method it is shown that particularly unfavorable combinations of operating conditions exist, within a range possibly encountered in railroad service, where thermally stable operation of the bearing cannot be obtained at high ambient air temperatures of about 100° F. Other examples show that increased bearing cooling and reduction or elimination of waste-pack friction provide significant decrease in bearing operating temperature and reduce the extent of the unstable operating regions. Reduction of oil viscosity, while also reducing the normal operating bearing temperature, increases the extent of the unstable region where bearing failure can potentially occur.

INTRODUCTION

In the solution of journal bearing problems it is normally possible, by analytical or experimental methods, to determine the principal performance characteristics of a bearing in terms of the operating parameters included in the Sommerfeld number. Such a solution, however, remains incomplete in practical cases where the oil temperature is not controlled. In such cases the oil viscosity is still treated as an independent variable in computation of the Sommerfeld number when, in reality, the oil viscosity existing under any specific condition is dependent upon all parameters which affect the oil temperature. Thus, the actual viscosity is dependent upon other factors determining the Sommerfeld number, upon the viscosity-temperature characteristics of the oil utilized, and upon the heat transfer properties of the particular bearing assembly. Inclusion of these various factors in the analysis introduces excessive complication and usually is not attempted.

In the analytical and experimental study of railroad freight car hot-boxes, determination of the actual operating temperature of the railroad journal bearings and the corresponding oil viscosity for a wide variety of practical operating conditions was essential. Such data could not be readily obtained from direct experiment on equipment in service because of the large number of variables involved and the difficulty of measuring and controlling many of the variables in field tests. As a substitute for such direct measurement, a method of analysis was established in this investigation to permit the determination of equilibrium bearing temperatures for any practical combination of railroad bearing operating conditions. The method also is used to show the effect of variations of each independent operating parameter upon the equilibrium bearing temperature.

The method of analysis used for determination of operating temperatures of the railroad journal bearing is described in the following sections and is a principal purpose of the present paper. In addition, the variation of equilibrium operating temperature of the railroad bearing is shown for different typical conditions.

BASIC DATA AND PRINCIPLES

An equilibrium value of average journal and bearing surface temperature (assumed, for simplicity, to be equal to the average oil film temperature in the bearing) is reached in the railroad journal bearing under any practical operating condition after sufficient time is allowed for the temperature (and oil viscosity) to adjust in value until the total friction heat generation is balanced exactly by the total heat dissipation to the atmosphere. This condition, and the corresponding equilibrium temperature, can be determined by analysis if all individual factors influencing friction heat generation and heat dissipation are known.

The method of analysis which is herein described is based upon such procedure. Information first is established to show the manner in which each of the important factors influencing friction heat generation and heat dissipation is affected over a broad range of load, speed, bearing fit conditions, and other basic variables. These data then are combined graphically to determine equilibrium conditions for heat flow.

Before discussion of the method used for graphical analysis of the data, it is required to specify the important factors which influence friction heat generation and heat dissipation in the railroad journal bearing assembly. Such factors include the following:

(a) Friction characteristics of the railroad bearing (120 degree centrally-loaded partial jour-

nal bearing) for various conditions of babbitt wear and bearing clearance

(b) Friction characteristics of the waste-pack lubricator

(c) Heat dissipation of the railroad journal box and truck assembly

(d) Viscosity-temperature characteristics of specific oil blends used in the railroad journal bearing.

These factors are discussed herein from the standpoint of the manner in which each of the factors is dependent upon basic operating parameters.

In further discussion given in this paper, the bearing oil-film temperature will be considered to be uniform and equal to an assumed uniform and average surface temperature of the bearing and journal. Actually, a considerable difference in temperature exists in the oil film, bearing, and journal, particularly along the axial length of the bearing assembly. Neglect of such variation permits simplification in the present analysis and does not seriously influence the significance and validity of the results.

Bearing Friction Characteristics

The friction characteristics for a 6 in. by 11 in, railroad bearing, expressed in terms of the coefficient of journal friction, are presented in a separate paper (1). Figure 1 is taken from this paper, and shows the variation of coefficient of journal friction with the commonly used parameter ZN/P, where Z is the oil viscosity in centipoise, N is the journal rotation speed in rpm, and P is the unit loading of the bearing in pounds per square inch. Such variation is given for several bearing fit conditions, including use of a new bearing with 0.033 in. radial clearance and use of a seated bearing. From the data in Fig. 1, it is evident that the journal friction horsepower can be computed for any selected operating condition, if specific values of oil viscosity are assumed.

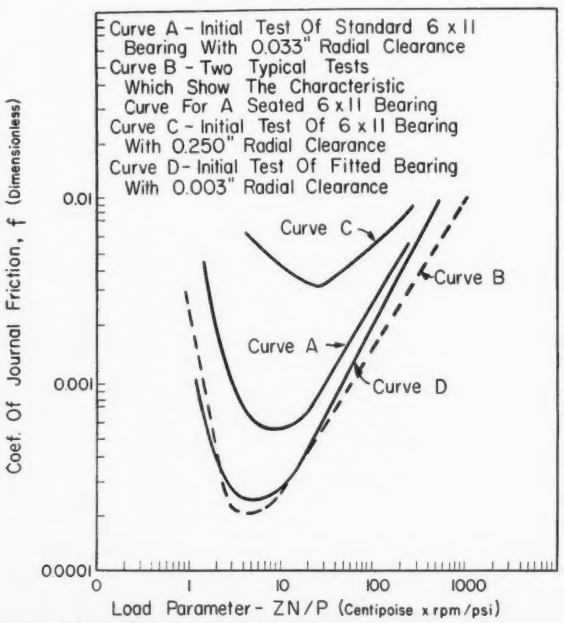
Curve A - Initial Test Of Standard 6 x 11 Bearing With 0.033" Radial Clearance
Curve B - Two Typical Tests Which Show The Characteristic Curve For A Seated 6 x 11 Bearing
Curve C - Initial Test Of 6 x 11 Bearing With 0.250" Radial Clearance
Curve D - Initial Test Of Fitted Bearing With 0.003" Radial Clearance
Coef. Of Journal Friction, f (Dimensionless)
0.01
0.001
0.0001
Curve C
Curve B
Curve A
Curve D
0
1
10
100
1000
Load Parameter - ZN/P (Centipoise x rpm/psi)

Figure 1. Variation of journal friction with ZN/P for four conditions of bearing clearance and wear condition.

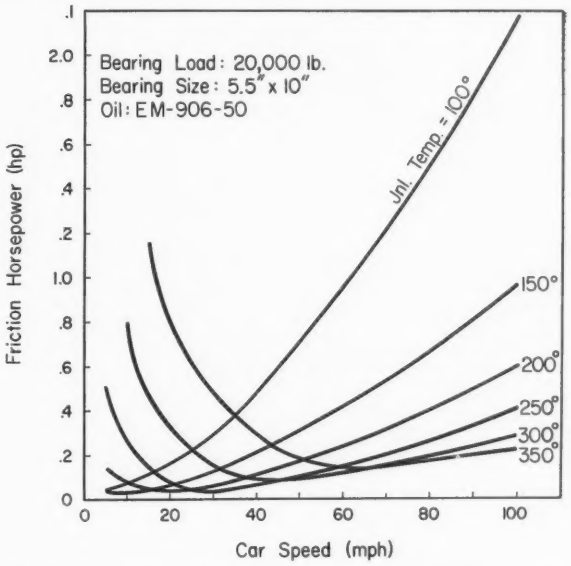

Figure 2. Friction horsepower (bearing only) for a well-seated railroad journal bearing.

The friction data of Fig. 1 are replotted in Fig. 2 to show the rate of bearing friction heat generation for particular operating conditions. In order to obtain Fig. 2, selection has been made of bearing load, bearing fit condition, and viscosity-temperature characteristics of the oil, and also particular values of average journal surface temperature (or oil film temperature) have been assumed. Curves similar to Fig. 2 may be established from Fig. 1 for any other selected combination of practical operating conditions.

The viscosity characteristics of the specific oil type (EM 906-50) used to establish Fig. 2, and of another oil type (EX-50) used in later analysis, are described below.

1. *EM 906-50 Oil*

This oil typifies those which meet AAR Specification EM 906-50 for car journal oil. This specification requires a viscosity of 50-55 sus at 210°F, and is normally met with a viscosity index (VI) of about 80.

2. *EX-50 Oil*

This oil is similar to the light car oil used experimentally in railroad journal bearing service. This type of oil has a viscosity of 40-42 sus at 210°F and normally has a VI value of about 80 to 90.

Waste Pack Friction Characteristics

The journal friction resulting from contact with the oil-saturated waste pack has been determined for various practical railroad operating conditions in extensive laboratory tests, described in reference (2). This study has shown that the coefficient of journal friction for the waste pack is a function of the waste pack pressure on the journal (P_w, in lb/in.^{2}), the oil viscosity (μ, in reyns), and the journal rotational speed (N', in rps). These data were obtained for a 5-1/2-in. journal, but can be applied with acceptable accuracy to other standard sizes of railroad journals for all conditions of cotton or wool waste, or other type materials sometimes used in railroad practice.

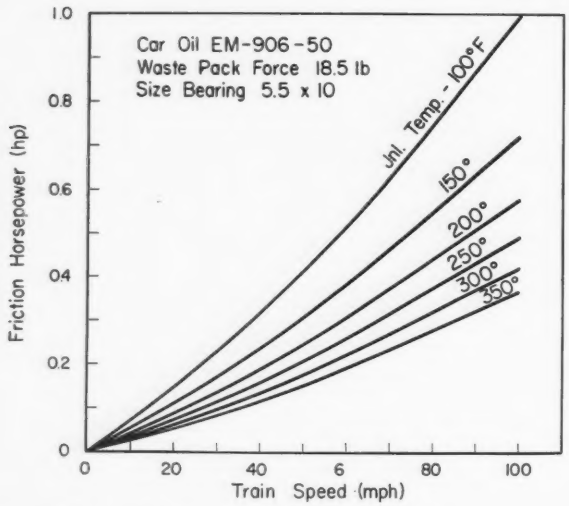

Figure 3. Typical variation of waste friction horsepower with train speed.

Values of waste pack pressure on the journal also have been measured and are known to vary over a wide range in practice corresponding to total force values of about 2 to 150 lb. The specific value obtained depends upon the weight of waste in the pack, the amount of new waste in contrast to used or renovated waste, and the amount of waste pack run-in or stabilization time after the journal box has been repacked.

The data of reference (2) are replotted in Fig. 3 to show the rate of heat generation associated with waste pack friction for specific selected conditions. In order to obtain Fig. 3, a particular oil and a particular value of waste pack pressure on the journal were chosen. Specific average oil film temperatures (and corresponding oil viscosity) also were assumed. Curves similar to those given in Fig. 3 may be obtained by a similar process for any other selected condition.

The data of Fig. 3 are in a form similar to that used in Fig. 2 for the heat generation associated with bearing friction. This form of data presentation is useful in later analysis.

Heat Dissipation Characteristics of the Journal Box and Truck Assembly

The heat dissipation characteristics of the standard railroad journal box and truck assembly, and of various modifications of this assembly, have been measured in the laboratory on a full-scale railroad truck with controlled air flow over the truck assembly. Such data are as yet unpublished. Typical results, applying to the standard journal box assembly, are presented in Fig. 4.

The curves given in Fig. 4 show the rate of heat dissipation (expressed in horsepower) obtained from the standard journal box and truck assembly for different car speeds, and with various average journal surface temperatures. The average journal surface temperature is expressed in terms of the temperature excess above that of the ambient air. As previously mentioned, it further is assumed to be equivalent to the average oil film temperature in the bearing.

The journal box cooling characteristics presented in Fig. 4 have been shown to vary to a small extent depending upon the area of contact between the components of the bearing and wedge assembly in the top of the journal box. The cooling characteristics also may be expected to vary to some extent with factors which influence the velocity of air flow over the truck assembly, such as the type of freight car and the location of the journal box on the car. The effect of such latter factors has not been measured, but may be estimated to have small magnitude.

The form of presentation in Fig. 4 is similar to that used in Figs. 2 and 3 in connection with bearing and waste friction. This form is used for later convenience in analysis. It is understood, however, that Fig. 4 refers to rate of heat dissipation, whereas Figs. 2 and 3 refer to rate of heat generation. Also, a specific value of ambient air temperature must be selected for any particular application in order to relate the temperatures shown in Fig. 4 to specific oil film temperatures.

METHOD OF ANALYSIS

The method used for determining equilibrium operating temperatures of the railroad journal bearing consists essentially of graphical comparison of factors which determine heat generation and heat dissipation. This comparison then permits a temperature to be determined which, for the selected operating conditions, corresponds to a balance between the heat generation in the journal box and the heat flow from the journal box to the cooling air.

The individual steps used in this analysis are outlined as follows:

Total Heat Generation

The total heat generated in the railroad journal box is associated essentially with two sources which are (a) the journal bearing friction, and (b) the friction of the waste pack rubbing on the journal. Specific values of heat generation from these sources have been presented in Figs. 2 and 3 for particular bearing operating conditions.

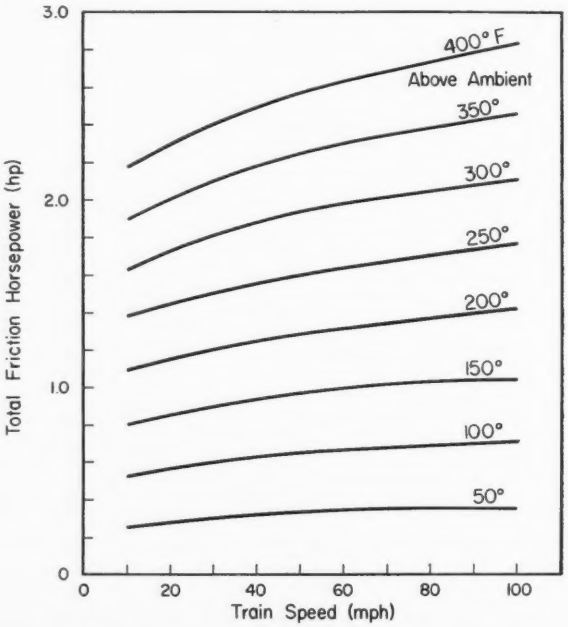

Figure 4. Cooling capacity of freight car truck for different car velocities and average journal temperatures above ambient, for 5½ x 10 bearing.

The total heat generation in the journal box is obtained by direct addition of these two factors. For the cases illustrated by Figs. 2 and 3, the total heat generation is shown in Fig. 5.

The curves of Fig. 5 thus show the variation of total heat generation in the journal box with car speed for the particular selected operating conditions and at various arbitrarily selected levels of oil film temperature. The particular oil film temperatures assumed are not important, as the immediate purpose is to establish a family of temperature curves covering a practical range of bearing operating conditions.

Curves similar to those shown in Fig. 5 may be established from Fig. 1 and from reference (2) for any other selected combination of operating conditions. Thus, each set of curves similar to Fig. 5 will represent a specific selected combination of the following factors:

(a) Bearing load
(b) Bearing clearance and wear condition
(c) Oil type
(d) Waste pack pressure

Equilibrium Rate of Heat Flow

Equilibrium conditions for heat flow are obtained by combined use of Figs. 4 and 5. In this procedure it is necessary to select particular values of ambient air temperature for use with Fig. 4.

The specific procedure utilized consists of superposition of the two sets of curves given in Figs. 4 and 5 and marking intersection points of the heat generation and heat dissipation curves which correspond to identical oil film temperatures. For example, each intersection of the 200°F curve of Fig. 5 (after correction for ambient temperature) with the 200°F curve of Fig. 4 provides a value of car speed and rate of heat flow which corresponds to equilibrium temperature conditions(where heat dissipation is equal to heat generation).

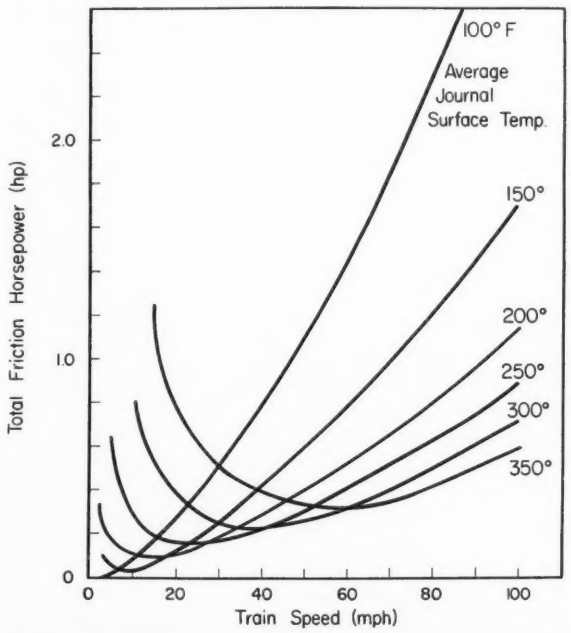

Figure 5. Typical friction horsepower curves for fully loaded bearing including waste friction (Same conditions as Figs. 2 and 3).

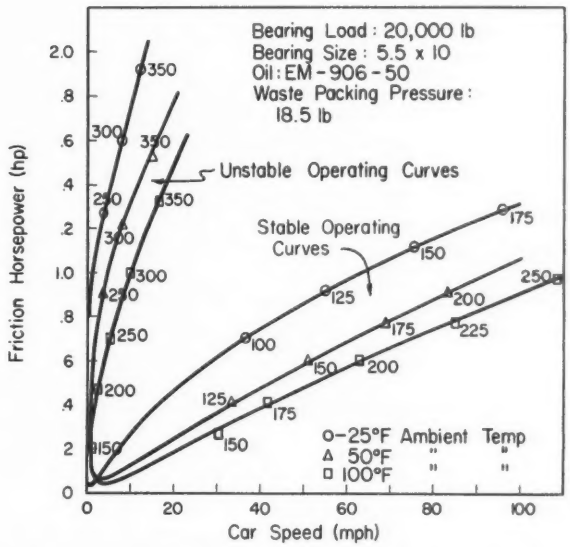

Figure 6. Total friction horsepower for seated bearing and waste under equilibrium temperature conditions.

Results of such procedure are illustrated in Fig. 6. In Fig. 6 the intersection points obtained from Figs. 4 and 5 are shown with the corresponding average oil film temperatures indicated, and each group of points corresponding to the same selected ambient air temperature are connected by a smooth curve. This curve shows the variation of total friction horsepower of the bearing and waste pack with car speed, for equilibrium bearing temperature conditions at the given ambient air temperature.

Equilibrium Temperatures

The curves presented in Fig. 6, while useful in showing friction power requirements, do not reveal the variations of equilibrium bearing temperature in a clear manner. For this purpose the data obtained for each point in Fig. 6 are replotted in different form, as shown in Fig. 7. The curves of Fig. 7 show directly the variation of the average oil film temperature with car speed for equilibrium conditions. Such equilibrium is reached after sufficient time of operation at the given car speed so that temperature stabilization is secured.

A dashed curve designated "critical temperature curve" also is shown in Fig. 7. This dashed curve does not signify equilibrium temperature conditions as the other curves in Fig. 7, but rather indicates the speed and temperature combinations where minimum bearing friction occurs (on the friction curve of Fig. 1) for the given operating conditions. The area above this curve in Fig. 7 corresponds to operation on the rising portion of the bearing friction curve (Fig. 1) under conditions of increasing oil film breakdown.

GENERAL DISCUSSION OF EQUILIBRIUM TEMPERATURE CURVES

The curves of Fig. 7, or similar curves derived for other selected operating conditions, show the average oil film temperature which will be obtained after temperature equilibrium is reached during service operation. The form of these curves, however, reveals additional information of considerable practical interest.

It is shown in Fig. 7 that the equilibrium temperature curve which is obtained for a given value of ambient air temperature has two branches. Thus, at a given car speed, two values of equilibrium temperature may be obtained. Closer examination reveals the significance of this result.

It is found that the lower branch of each curve given in Fig. 7 is formed, during the graphical procedure, from the branch of the bearing friction curve (Fig. 1) which corresponds to a full developed hydrodynamic lubricating film. The upper branch of each curve in Fig. 7 is formed from the branch of the bearing friction curve (Fig. 1) which corresponds to rapidly rising friction in the region of oil film breakdown. Further analysis shows that the lower branch of each curve in Fig. 7 represents conditions of stable temperature equilibrium, such that any transient deviation from equilibrium conditions will be followed by a return to the condition of stable heat balance indicated by the curve. The upper branch of each curve in Fig. 7, however, represents an unstable equilibrium condition. Any transient deviation from an operating condition lying on this curve will be followed by increased deviation. For example, if an operating speed and temperature combination lying above this upper branch should be reached in any manner, a tendency will exist for the bearing temperature to continue increasing in an unstable and uncontrolled manner. For transient deviations below the upper branch of the curve in Fig. 7, a tendency exists for the temperature to continuously decrease until stable equilibrium is reached on the lower branch of the curve.

The interpretation of the temperature equilibrium curves given above is found from detailed study of the indicated direction of heat unbalance on each side of the curve branches. The result signifies physically that, if bearing operation is extended into the region of oil film breakdown and rising friction (illustrated in Fig. 1), a point is reached where temperature instability is obtained. The position of this point on the curve of Fig. 1 is dependent upon the existing heat dissipation, car speed, and other factors and is defined quantitatively by the upper branch of the curve in Fig. 7.

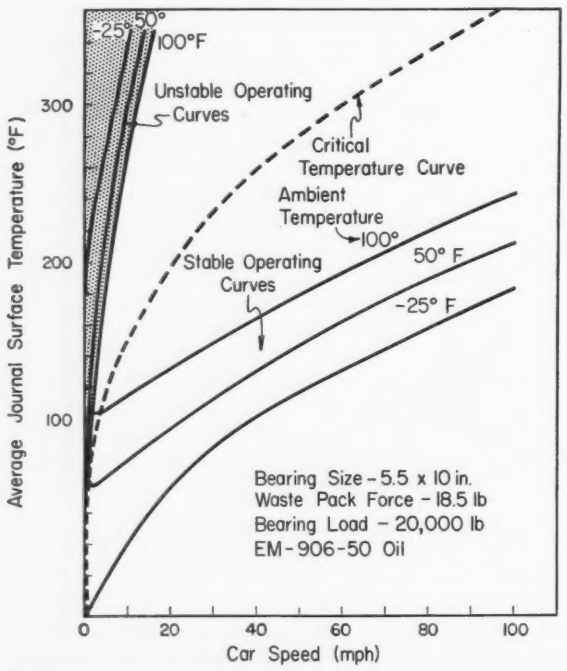

Figure 7. Variation of equilibrium bearing temperature with car speed and ambient air temperature for particular conditions.

As the portion of the bearing friction curve (Fig. 1) corresponding to oil film breakdown can be defined experimentally with less accuracy than the portion of the curve corresponding to full hydrodynamic oil film conditions, the location of the upper branch of the temperature equilibrium curve in Fig. 7 also is somewhat less accurately defined than the position of the lower branch. However, moderate or large changes in position of the upper branch of the equilibrium curve in Fig. 7 (as shown later to be obtained for other bearing operating conditions) are significant.

The general interpretation of the temperature equilibrium curves presented in Fig. 7, as discussed above, is summarized in the following manner. The lower branch of the curves applies to most practical operating conditions of the railroad journal bearing. It shows, within the accuracy of the data and analytical methods used, the average oil film temperature which will be obtained for the operating condition specified after sufficient operating time is allowed for temperature equilibrium to be obtained.

Under transient operating conditions, such as during and immediately following a change in car speed, the transient speed and temperature combination will not fall upon the stable equilibrium curve. In general, however, after sufficient time again has been allowed for temperature equilibrium to be reached for the new operating condition, the operating point will return to the stable equilibrium curve (lower branch in Fig. 7).

If, during such transient changes, a temperature and speed combination should be reached which lies above the unstable equilibrium temperature curve (upper branch in Fig. 7), a tendency then will exist for the temperature to rise in an uncontrolled manner if such speed is maintained. The practical seriousness of such an effect is not brought out strongly by Fig. 7, where the bearing operating conditions represented are relatively favorable and the area above the upper branch of the curve is small. It will be shown later that this effect assumes greater practical importance where the bearing operating conditions are less favorable.

It is noted that a third branch of the temperature equilibrium curves presented in Fig. 7 also exists, although it is not shown. This third branch, which has small practical interest, lies close to the vertical coordinate axis of Fig. 7. It arises from the form of the total heat generation curves presented in Fig. 5 which must return to the coordinate origin at zero car speed, although this again is not shown in Fig. 5. The added portion of the curve in Fig. 5 would provide an additional intersection point at each temperature when Figs. 4 and 5 are combined graphically to form Figs. 6 and 7. The third branch of the curve physically represents full boundary friction conditions in the bearing where the coefficient of friction is essentially independent of load and speed and is only moderately dependent upon bearing temperature.

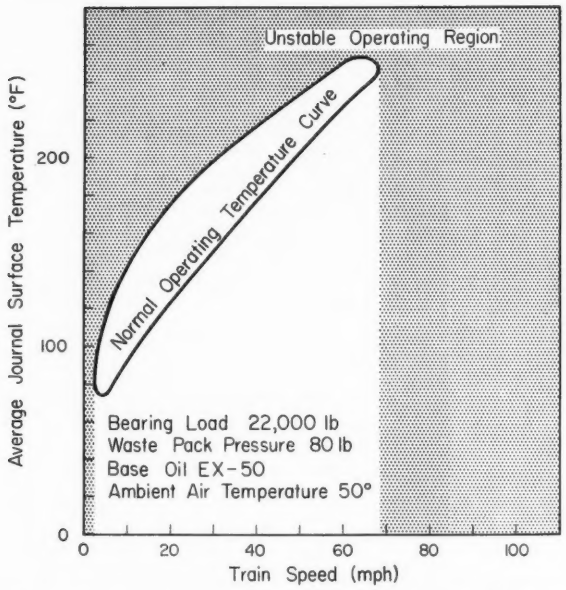

Figure 8. Performance curve for a new, unworn 5½ x 10 bearing on a full-size journal.

APPLICATION OF ANALYSIS TO TYPICAL BEARING OPERATING CONDITIONS

The method of analysis previously described may be applied to any selected combination of railroad bearing operating conditions to determine (a) the resulting stable equilibrium temperature of the bearing, and (b) the extent of the unstable operating region where uncontrolled rise of bearing temperature is encountered. The effect of possible or proposed changes in the bearing arrangement also may be investigated in a similar manner to evaluate such factors as changes in oil type, removal of waste pack friction, and improvement in bearing cooling. The following discussion presents typical applications of the method of analysis to such cases for further illustration.

The temperature equilibrium curves presented in Fig. 7 represent, as previously mentioned, reasonably favorable operating conditions for a 5-1/2 in. x 10 in. railroad bearing. In contrast, Fig. 8 shows the equilibrium temperature curve (for 50°F ambient air temperature) obtained under a less favorable combination of operating conditions. In the case illustrated by Fig. 8 (as compared to Fig. 7), the bearing load has been increased (from 20,000 lb. to 22,000 lb.), the waste pack force on the journal has been increased (from 18.5 lb. to 80 lb.), a less viscous oil (EX-50 oil) is used, and a new bearing (1/32 in. radial clearance) is utilized instead of the seated bearing considered in Fig. 7.

It is shown in Fig. 8 that the two branches of the equilibrium temperature curve (for 50°F ambient air temperature) have moved closer together as compared to Fig. 7 and now form a closed loop. The significance of the two branches of the loop remains identical to that discussed in connection with Fig. 7. It is concluded, therefore, that the bearing will operate in a stable manner only in the unshaded area on Fig. 8, and, after any transient operation within this unshaded region, the temperature will stabilize at a value on the lower branch of the loop. Transient operation into the shaded portion of Fig. 8 will result in uncontrolled increase of bearing temperature.

As a specific example in Fig. 8, operation at 60 mph will result in an equilibrium value of bearing temperature of about 224°F. If the speed then is decreased at such rapid rate that the bearing temperature remains essentially constant, the bearing temperature will start to increase indefinitely during further sustained operation at any speed less than about 45 mph.

With further reference to Fig. 8, it is found that when an ambient air temperature of 100°F is assumed, it is not possible to establish an equilibrium temperature curve for the given conditions. This result indicates that, with such increased ambient air temperature, the bearing cannot operate under the given conditions in a stable manner at any speed.

For the practical conditions illustrated by Fig. 8, it is understood that the bearing performance will improve beyond that shown, possibly at a rapid initial rate, as the new bearing wears and becomes seated on the journal.

The equilibrium temperature curve given in Fig. 8 is obtained for a railroad journal bearing having the normal heat dissipation characteristics described previously in Fig. 4. It is of interest to investigate, by means of the analytical method described, the effect of increasing heat dissipation from the bearing. This procedure can be accomplished, for example, by dividing in half each of the temperatures shown in Fig. 4 to produce an arbitrary hypothetical condition where heat dissipation from the bearing has been doubled. Practical means for accomplishing this condition are not considered for the present illustrative purpose.

The foregoing procedure has been applied to obtain the equilibrium temperature curve shown in Fig. 9

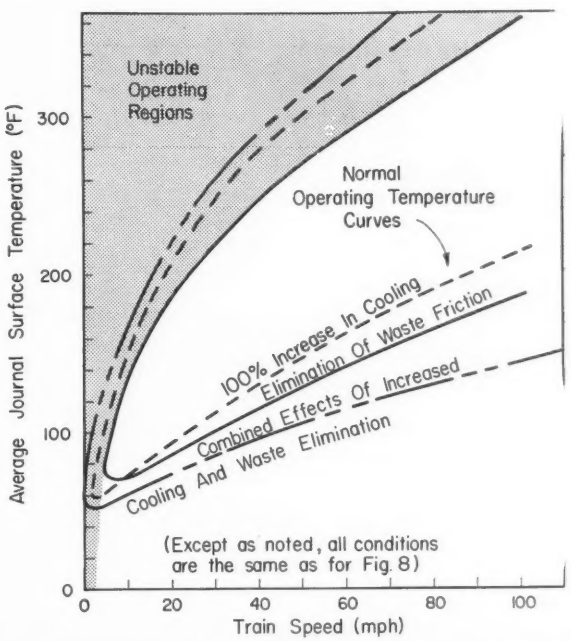

Figure 9. Journal bearing performance curves showing effect of eliminating waste friction and improving turck cooling capacity by 100 percent.

for 100 per cent increase in bearing cooling. This curve should be compared to the curve of Fig. 8, where all assumed conditions are identical except for the bearing cooling properties.

There is presented also in Fig. 9 an equilibrium temperature curve representing a condition where the waste-pack friction has been eliminated, such as by use in practice of a lubricator with small friction. This curve was obtained by considering the bearing friction to constitute the sole source of heat generation, and by eliminating the waste-pack friction curves (similar to Fig. 3) from the analysis. This curve in Fig. 9 also should be compared to the curve of Fig. 8, where all assumed conditions (including rate of cooling) are identical except for the waste-pack friction.

There is shown also in Fig. 9 an equilibrium temperature curve for the condition where both 100 percent increase of heat dissipation (compared to the normal railroad bearing assembly) and elimination of waste-pack friction are assumed.

The equilibrium temperature curves of Fig. 9, when compared to Fig. 8, show that increased bearing cooling and elimination of waste-pack friction have similar effects upon bearing performance when applied separately. In each case, equilibrium temperatures obtained during stable operation are decreased, and the area of unstable operation is reduced. When both factors are applied simultaneously, further gain in both respects is obtained.

An additional example of application of the described method of analysis is shown in Fig. 10. The dashed curve shown in Fig. 10 is identical to the equilibrium curve for 50°F ambient air temperature given previously in Fig. 7, as all assumed conditions are identical in the two cases. The solid curve of Fig. 10 also represents the same operating conditions except for the use of EX-50 oil having lower viscosity than the EM 906-50 oil utilized in the previous case.

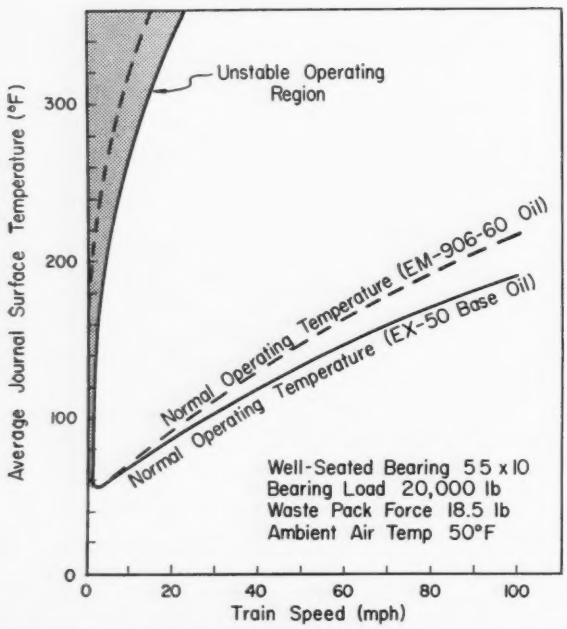

Figure 10. Journal bearing performance curves showing effect of oil viscosity.

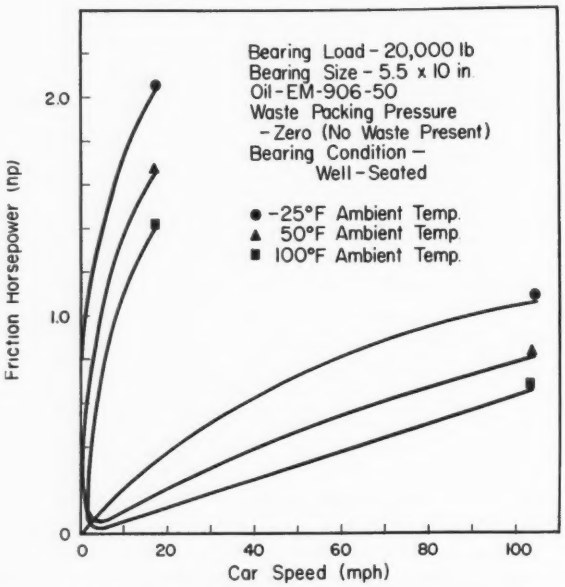

Figure 11. Total friction horsepower under equilibrium temperature conditions with waste pack friction eliminated.

It is shown in Fig. 10 that the general effect of using less viscous oil is to reduce the equilibrium oil film temperature during normal stable operation of the bearing, but simultaneously to increase the extent of the operating region in which temperature instability occurs. The latter effect is not serious for the particular operating conditions selected for Fig. 10, but may be more serious under severe conditions of bearing operation.

A general comparison of Figs. 9 and 10 shows that use of less viscous oil has an effect similar to that of increased bearing cooling (or reduced waste pack friction) with respect to providing reduction of normal equilibrium operating temperature of the bearing. An opposite effect is obtained in the two cases, however, with regard to the extent of the unstable operating region, which is related closely to degree of bearing operating safety.

In the foregoing discussion principal emphasis has been placed upon the equilibrium temperature curves and the extent of the unstable operating region of the bearing. While such factors appear to hold principal interest in analysis of bearing operation, consideration of the total friction power requirement of the bearing and waste pack for the equilibrium temperature conditions obtained in service may also have practical importance. Such data are obtained during the analytical procedure described, as already mentioned and as shown in Fig. 6.

An additional example of friction power requirements under equilibrium temperature conditions is given in Fig. 11. The operating conditions selected as a basis for Fig. 11 are identical to those used for Fig. 6, except that Fig. 11 is based upon the assumed condition that waste pack friction is eliminated (such as through use of a lubricator having small friction). Data of Fig. 6 were based upon a total pressure force of 18.5 lb. between the waste pack and journal.

Examination and comparison of Figs. 6 and 11 show that the total power required to overcome friction under equilibrium operating conditions varies from about 0.4 hp to 1.0 hp at 60 mph for the specific conditions

illustrated. The friction power increases with car speed and with decrease in ambient air temperature. Elimination of waste pack friction results in considerable decrease in total friction power.

Friction power curves similar to Figs. 6 and 11 may be obtained for any practical combination of railroad bearing operating conditions through application of the analytical method described.

CONCLUSIONS AND SUMMARY

The graphical method of analysis, which has been described for determination of equilibrium values of the average oil film temperature in railroad journal bearings, is useful for investigating the effect of numerous variables which affect bearing operation in service. The method is relatively easy to apply, and appears to provide reasonable accuracy consistent with that of the laboratory data upon which the procedure is based. Although no check is available upon the absolute accuracy of the results because of lack of field data based upon closely controlled conditions, such field data which exist appear to be generally consistent with the results obtained.

This method of analysis has been found to reveal regions of bearing operation in which temperature instability occurs, or where continuous and uncontrolled increase of bearing temperature is obtained. Although basically this concept is not new (3) and normally is associated with journal bearing operation under a condition of oil film breakdown, the analytical method which is utilized provides convenient presentation of such information in a form which can be easily obtained and interpreted and which includes all main factors relating to heat generation and heat dissipation in the bearing assembly.

Use of the method in analysis of railroad journal bearings has been found to provide the only practical manner in which extensive information regarding bearing operating temperatures can be secured. The conducting of numerous laboratory tests with elaborate full-scale equipment or the conducting of extended field tests with close control of all significant variables, would otherwise be required. Such tests would be expensive and time consuming.

Application of the analytical method to particular operating conditions of the railroad journal bearing provides the following general conclusions:

(1) Specific conditions are found, within the range of conditions which may be encountered in railroad service, where the railroad journal bearing will not operate without an uncontrolled continuous increase of temperature. However, such circumstances are found only for the most unfavorable combinations of the numerous operating variables, including use of a new unfitted bearing, large waste pack pressure, large bearing load, high ambient air temperature, and low viscosity oil.

(2) The reduction or elimination of waste pack friction and possible improvement of bearing heat dissipation provide similar effects in reducing equilibrium operating temperatures of the bearing and also in reducing the extent of the unstable operating region.

(3) Increase of oil viscosity generally tends to increase the equilibrium operating temperature of the bearing, but reduces the extent of the unstable operating region.

(4) Friction power associated with normal operation of the railroad journal bearing assembly is reduced significantly with elimination of waste-pack friction.

ACKNOWLEDGEMENT

The research program upon which this paper is based was carried out by the Armour Research Foundation of the Illinois Institute of Technology and was sponsored by the Association of American Railroads.

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Selected literature compiled by members of the Technical Committees and Industry Councils of the Society

Literature Abstracts —— W. E. Campbell, Editor

"The Relation Between Saybolt Universal Viscosities and Kinematic Viscosities." Masumi Kinoshita (Shinagawa Oil Research Laboratory, Showa Oil Co., Tokyo, Japan) *J. Inst. Petroleum, v. 43, 1957, 161-4.*

It is found that the following empirical equation, which is of the same type as the equation previously deduced by the author to express the relation between Redwood seconds and kinematic viscosities, is also suitable for the conversion of kinematic viscosities to Saybolt Universal viscosities in the whole viscosity range greater than about 2 centistokes (about 33 Saybolt Universal sec.) at temperatures between approximately 70° and 300°F.

$$S = (4.6050 + 0.000297t) \div \left(1 - 10^{-0.07445\nu^{0.9538}} \right)$$

where S and ν are Saybolt Universal viscosities in seconds and kinematic viscosities in centistokes respectively, and t is temuerature, °F. *(Abstractor: E. O. Forster)*

The Relation Between Redwood No. 1 Seconds and Kinematic Viscosities." Masumi Kinoshita (Shinagawa Research Laboratory, Showa Oil Co.,) Tokyo, Japan. *J. Inst. Petroleum, v. 43, 1957, 152-60.*

The relation between Redwood seconds and kinematic viscosities is studied, the experimental data obtained on four conventional Redwood No. 1 viscometers being compared with those on a set comprising five suspended-level master viscometers with long capillary tubes of different diameter. Twenty-seven oil samples of suitable viscosities covering the range 2-200 centistokes at temperatures between 30° and 100°C. are used. Statistical analysis of the experimental data and derived values indicate that

$$\log \log [A \div (A - \nu/R)] = \log k + n \log \nu$$

where R and ν are Redwood seconds and kinematic viscosities respectively, A is a value of ν/R in the viscosity range greater than about *70* centistotes and k and n are constants independent of both temperature and correction factors of Redwood viscometers. Furthermore, it is indicated that the reciprocal value of A, or $1/A$, has a linear relation to temperature t, as in the following equation

$$1/A = a_0 + a_1 t$$

where, a_0 and a_1 are constants.

Research work published by other investigators, such as G. Barr and F. H. Garner et al., is also discussed, together with experimental results by the author. It can be emphasized that the following empirical equation, which is obtained by combining the above two equations, can be used successfully for the conversion of Kinematic viscosities to Redwood No. 1 seconds in the whole viscosity range greater than about 3 centistotes (about 34 Redwood sec.) at the temperatures between 20° and 100°C. approximately.

$$R = (4.048 + 0.00062t)\ \nu \div \left(1 - 10^{-0.06668\nu^{0.9704}} \right)$$

where R and ν are Redwood No. 1 seconds and kinematic viscosities in centistokes respectively, and t is temperature, °C. Furthermore, it is found that Ubbelohde's conversion table, which is currently used for the conversion of kinematic viscosities to Redwood No. 1 seconds, possibly represents the relation between kinematic viscosities and Redwood seconds obtainable in the Redwood viscometer No. 1307, which is authorized as the standard by the Institute of Petroleum, with a deviation of about 0.5 percent; however, the table does not represent with certainty the relation obtainable in other Redwood viscometers, such as No. 3967 and other commercial instruments used by the investigators mentioned above, as well as by the author. The Redwood seconds obtained by conversion from kinematic viscosities by means of the above-mentioned table show a characteristic deviation from the values observed in the Redwood instrument at the range of about 30-120 sec. (2-30 centistokes), and the deviation reaches a maximum amounting to as much as 3 per cent or more in the limited range of about 40-60 sec. (6-12 centistokes). *(Abstractor: E. O. Forster)*

"Radiochemical Method for Investigating the Stability of Additive Solutions in Lubricating Oils" by Yu. S. Zaslavski, R. N. Shneerova and G. I. Shor. (in Russian) Zavodskaya Lab., *22*, no. 4, 1956, 417-8. *A.T.S. Translation RJ-579.*

A main objective of this study is the determination by radiochemical means of the stability of additive solutions in oil under the influence of added water. The additive containing a β—radioactive isotope is compounded with several base oils. Radioactivity is measured by an end counter of the type TM-20, placed in a lead chamber. The calibration curves for each of the radioactive oils is first determined under identical geometrical conditions of the sample container. The radioactivity of the oil sample is converted to the weight concentration of the additive using the calibration curve or an appropriate conversion coefficient. A correction for decay is made. The radioactivity of 15 samples of each oil is measured to determine the average amount of additive originally in the oil. Samples with no water and with 20 per cent of water are centrifuged and the average amount of additive in the oil is determined by radioactive measurement of the centrifuged oil. Table 1 gives results characteristic of the effect of water content on the stability of solutions of the additive AzNII-4 in various oils.

Table 1
The Effect of Water on the Stability of the Additive Solution

	Separation of additive, %	
Oil	Without Water	20% Water in Oil
SU + 3% AzNII-4	0.3	3.8
Diesel GOST 5304-54 + 3% AzNII-4	1.0	11.0
AS-9.5 + 3% AzII-4	1.1	9.6

(Abstractor: E. H. Loeser)

"The Effect of Temperature on the Boundary Lubrication. Friction Coefficients and Surface Potentials of the Films of Mono- and Multi-molecular Layers," T. Isemura and R. Nakagawa. *Memoirs of the Institute of Scientific and Industrial Research.* Osaka University (Japan) *v. 12, 1955, 129-136.*

Static friction is measured at various temperatures with an apparatus which pulls a convex glass slider over a chromium plated brass or a glass plate. The plate is prelubricated by mono- or multi-molecular layers of stearic acid deposited by the Langmuir-Blodgett technique. An increase in temperature decreases friction with a monolayer but has no effect with multilayers. The *p*H of the substrate from which the film was built up has an effect if the plate is glass, but none if the plate is metal. A transition is observed at the melting point of stearic acid only if the system is acid. The temperature has a marked effect on surface potential. Boundary lubrication is discussed in terms of the lateral cohesion of the organic molecules. *(Abstractor: Douglas Godfrey)*

Grease Flow in Shielded Bearings

(Continued from p. 107)

Thus, while the concept of the grease as an entity providing the lubrication is a valid one under these restricted conditions, the concept of a storehouse for oil has not been refuted. Another attempt to study this might be the application of similar techniques to open bearings with grease packed housings to see how far away from the bearing the soap is disturbed.

The discussor has been running tests for the last 3 years with radioactive tracers in the oil phase placed well away from the bearings at the back of greased housings. The active agent in the oil has slowly but certainly been working its way into the bearings. The rate is slow, but adequate for ordinary electric motors which may run 10 years or more without regreasing.

Even without radioactive techniques there is strong evidence that oil flow from a grease reservoir is an important factor in lubrication. Our laboratory has data from years of testing greases in open and sealed bearings which show that the extra grease well away from the moving parts is very beneficial. Even putting grease outside a sealed bearing where only the oil by leaking into the bearing can be effective, will add to the bearing life.

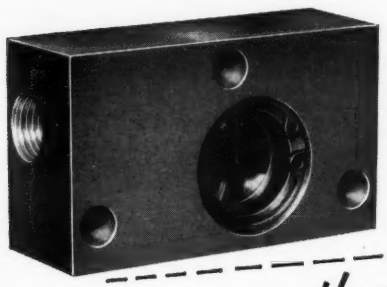
I would suggest that had these tests run long enough, redistribution of the oil would also be detected. There is little tendency for the oil to flow in the grease until some portion of the gel structure has been depleted by bleeding, evaporation or oxidation. I believe that is why, in the packed housing — open bearing system, it takes so long for the oil to move. It must wait for the oil nearest the bearing to be dissipated by leakage or evaporation. Then the reservoir can provide make-up oil. Lubrication failure occurs when the reservoir is as dry as the more active grease.

Nevertheless, whatever the mechanism is, or mechanisms are, the authors have certainly shown that no simplication should be uncritically accepted. How important even a little grease flow in the bearing may be is still a question. Their technique may be of invaluable assistance in settling that point.

AUTHOR'S CLOSURE

Dr. Jackson's comment to the effect that we have not proved that grease moves as an entity as a general rule for all bearings is certainly valid. We do feel that the shielded bearing type of application which we studied is an important one. For example, it is the type used as a standard for evaluating bearing life characteristics by many bearing manufacturers, grease technologists and the military. It is probably true that the relative amounts of flow in comparison to the amount of grease in the bearing assembly would vary markedly with bearing design, as well as speed of rotation and hardness of the grease which we demonstrated.

The radioactive tracer studies described by Dr. Jackson do not really shed light on whether or not the grease as an entity is moving. He showed that the oil migrated but had no measure of soap movement. Our point, of course, is that one should measure movement of both components, and particularly the soap migration, to determine the actual movement of grease. We hope that the technique described in our paper will facilitate work on defining actual movement, or lack of it, in a wide variety of grease applications.



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Patent Abstracts

Compiled by
Ann Burchick, Aluminum Co. of America

Lubricating Composition, Patent No. 2,774,733 (P. H. Williams and L. B. Scott, assignors to Shell Development Co.) A lubricating composition adapted for extreme pressure use comprising a major amount of a water-miscible mixture of monohydroxy oxyethylene oxy-1, 2-propylene aliphatic monoethers in which the ethylene oxide and 1,2-propylene oxide are combined therein as oxyethylene and oxy-1,2-propylene groups in a ratio which is at least one-third part of 1,2-propylene oxide for each part of ethylene oxide by weight, said mixture having an average molecular weight of at least 300 attributable to said groups, the aliphatic monoethers of said mixture containing in a single molecule both the oxyethylene and oxy-1,2-propylene groups, and a minor amount, sufficient to improve the extreme pressure properties thereof, of citric acid.

Oxidized Oil Product and Process for the Production Thereof, Patent No. 2,774,780 (J. J. Szabo and D. Frazier, assignors to The Standard Oil Co.) The method of forming a soluble oil which comprises oxidizing a solvent-extracted mineral oil having an aniline number of about 107 C. and a viscosity of about 115 SSU at 110 F. in an oxidation zone maintained at a temperature of about 320 F. and a pressure of about 57 lbs. per square inch gauge, continuously introducing said oil into the oxidation zone and continuously withdrawing the oxidized oil from said oxidation zone at substantially the same rate so that the oil has an average residence time in the oxidation zone of about 2 hours, said withdrawn oil comprising two immiscible fractions, the lighter of which has a saponification value of about 81 and the heavier of which has a saponification value of about 250, permitting the oxidized product to separate into two layers, separating the highly oxidized lower layer from the lesser oxidized upper layer, and neutralizing the highly oxidized lower layer with potassium hydroxide to form a soluble oil miscible with water as a stable emulsion.

Improved Lubricants, Patent No. 2,775,559 (C. M. Himel and L. O. Edmonds, assignors to Phillips Petroleum Co.) An improved lubricant comprising a major proportion of a mineral lubricating oil containing an alkylene polyamine partially N-substituted with at least one of RS-- and RSS-- in which R is a radical selected from the group consisting of alkyl, substituted alkyl, aryl, substituted aryl, aralkyl and substituted aralkyl, in an amount sufficient to impart antioxidant and anticorrosive characteristics to said lubricant, but less than 8 weight percent of said oil.

Corrosion Inhibiting Lubricating Compositions, Patent No. 2,775,560 (H. M. Lurton and L. C. Westcott, assignors to Shell Development Co.) A vapor phase and liquid phase corrosion inhibiting lubricating composition comprising 0.015% C_{18}-alkenylsuccinic acid, 0.5% 2,6-ditertiarybutyl-4-methyl phenol, 0.02% sulfurized oleic acid, 0.05% n-nonanoic acid and the balance being a mineral lubricating oil.

Lubricant Compositions, Patent No. 2,777,820 (D. A. Hirschler, Jr., assignor to Ethyl Corp.) An improved crankcase lubricant composition for spark ignition internal combustion engines consisting essentially of a crankcase lubricating oil and from about 3 to about 15 percent by weight based on the weight of said oil of acenaphthene.

Oxidation-Inhibited Mineral Oil Compositions, Patent No. 2,779,738 (J. J. McBride, Jr., assignor to Tidewater Oil Co.) A composition comprising a mineral oil of lubricating grade in major amount based on the weight of the composition and a small amount, sufficient to impart improved antioxidant characteristics to said oil, of dimethyl-bis-(p-dimethylaminophenyl)-silane.

Turbine Lubricating Oil Compositions, Patent No. 2,799,739 (J. D. Spivack, assignor to Tidewater Oil Co.) A lubricating composition comprising a mineral oil of turbine lubricating grade in major amount based on the weight of the composition having dissolved therein from about 0.01 to about 1.0 percent by weight of a combination additure of mono- and di-lauryl acid phostive consisting essentially of a mixphate and a fatty acid from the group consisting of caprylic acid and lauric acid in a weight ratio of one part of said mixture of phosphates to from about one to about five parts of the fatty acid.

Mineral Oil Lubricating Compositions, Patent No. 2,779,740 (N. V. Messina, assignor to Tidewater Oil Co.) A mineral oil lubricating composition comprising a mineral oil of lubricating grade in major amount based on the weight of the composition, a small amount, sufficient to impart rust- preventive properties to said composition, of an oil-soluble rust-inhibiting alkenylated succinic acid having a molecular weight of about 400, a small amount, sufficient to impart anti-oxidant properties to said composition, of an oil-soluble anti-oxidant phenyl naphthylamine, and a small amount, sufficient to inhibit said composition against formation of relatively stable emulsions with water, of an oil-soluble alkyl amine having from 4 to 18 carbon atoms.

Production of Lubricating Oils, Patent No. 2,780,581 (R. A. Macke and E. F. Wadley, assignors, by mesne assignments, to Esso Research & Engineering Co.) A combination process for producing lubricating oil comprising the steps of catalytically cracking a gas oil boiling within the range of about 650 to 1100 F., segregating a first oil fraction boiling within the range of about 650 to 1100 F. from the cracked products, segregating a second oil fraction boiling within the range of about 650 to 1100 F. from paraffinic crude oil, and contacting a mixture of said first and second oil fractions with a selective solvent for aromatic compounds to remove aromatic compounds from the said mixture, said mixture containing about 50% of said first oil fraction, said process including the step of dewaxing said oil fractions.

Lubricating Compositions, Patent No. 2,780,597 (P. H. Williams, L. B. Scott and V. N. Borsoff.) A lubricating composition adapted for extreme pressure use comprising a major amount of mineral lubricating oil having incorporated therein from 0.1 to 10% of a salt selected from the group consisting of sodium dihydrogen phosphate, potassium dihydrogen phosphate and ammonium dihydrogen phosphate, said salt being in colloidal dimensions of from about 1 to 10 microns.

Otto Cycle Engine Fuels and Lubricants Containing Halohydrocarbon, Patent No. 2,784,160 (R. H. Blaker, assignor to E. I. duPont de Nemours & Co.) A crankcase mineral lubricating oil for Otto cycle engines containing from about 2% to about 10% by weight

Section News

Pittsburgh — Section members heard J. O. McLean, ASLE President, speak on "Lubrication in the Aluminum Industry" at February meeting. March session featured discussion of "Quenching Oils" by W. J. Varetoni of Cities Service Oil Co. Highlight of April meeting will be tour of Gulf Oil Co.'s new engine laboratories. . . . **Rochester** — Speaker at February meeting, George Miller of Battenfield Oil & Grease Co., outlined "Characteristics and Uses of Various Grease Products." Those in attendance at March meeting heard talk on "Proper Selection of Preservation Materials" by Walter Spencer, Daubert Chemical Co. April 9 meeting will have members of Rochester-sponsored lubrication course on hand as dinner guests and recipients of course-completion certificates. . . . **Puget Sound** — Al Piovesan, design engineer for Boeing Aircraft, gave slide-illustrated talk on the Boeing gas turbine engine at January meeting. . . . **Houston** — Nominating committee was elected after February steak dinner at Ben Milam Hotel. Members later heard W. C. Woerner of Magnolia Petroleum Co. discuss "Care and Maintenance of Antifriction Bearings." . . . **Kansas City** — February meeting was visited by C. L. Willey and A. B. Wilder, ASLE midwestern regional vice president. Speaker was M. W. Brooks of the Cummins Co. of St. Louis, Mo. Officers will be elected at the March 24 meeting, and a talk on "A Workable Maintenance Program for Hydraulic Pumps, Motors, and Auxiliary Equipment" will be presented. Officers will be installed at the April 21 meeting. . . . **Twin Cities** — February dinner meeting was preceded by a tour of Continental Machines, Inc. of Savage, Minn. NLGI's movie, "Grease, the Magic Film" was shown. . . . **Buffalo** — "Grease Characteristics and Applications" was Sinclair Refining Co.'s George E. Kellis' subject at February meeting. . . . **Montreal** — Centralized lubricating systems were analyzed by February speaker Howard Marten. "Hydraulic Machinery" was subject of March talk by R. H. Krepps of American Brake Shoe Co. Slated for April is discussion of "Cost Reduction by means of a Lubrication Program" by C. L. Pope of Eastman Kodak. . . . Milwaukee — Zeno Le Tellier of Pate Oil Co.'s topic for March 20 meeting is "Facts on Petroleum." Tour of the A. O. Smith Corp, is planned for April 17 at 7 P. M. . . . **Los Angeles** — Dr. Peter A. Aseff, Lubrizol Corp., presented an up-to-date report on "Automotive Gear Lubrication to Meet New Demands." . . . **Chicago** — February 20 was Ladies Night with D. R. Mabon showing that it's a woman's world with his talk on "Magic: How Women Influence Chemical Progress." . . . **Intermountain** — New officers elected at February meeting. March meeting highlighted by Ford Motor Co.'s R. O. Kageff's talk on plant lubrication.

OBITUARY

Members of the American Society of Lubrication Engineers are saddened by the death of L. Thomas Hill, 48, of Toronto, Canada. Manager of the head office lubricants department of the Shell Oil Co. at the time of his death, Mr. Hill worked his way up through the ranks of that organization, beginning as a service station attendant in 1930. Three years later, he became a clerk in Shell's Toronto division office and in 1940 was promoted to a city salesman. Two years later, he was transferred to Thorold, Ont., as an industrial lubricants salesman. In the years that followed, Mr. Hill moved up through the lubricants department of company into positions of increasing responsibility.

In addition to his ASLE membership, Mr. Hill was a member of the Engineers' Club of Toronto and the Society of Automotive Engineers.

Patent Abstracts (cont.)

of at least one halogenated fluorine-containing ethane of the group consisting of 1-bromo-1, 1-dichloro-2,2,2-trifluoroethane, 1,2-dibromo-1-chloro-1,2,2-trifluoro-ethane and 1,1,2,2-tetrachloro-1,2-difluoroethane.

Protective Lubricant Composition, Patent No. 2,785,111 (E. R. Vierk and T. J. Karr, assignors to Sinclair Refining Co.) A protective lubricant containing about 5 to 20 percent by weight of microcrystalline wax melting in the range from about 140 to 165 F., about 50 to 70 percent by weight of blown asphalt, said blown asphalt being obtained by air blowing asphalt having a viscosity of about 900 to 1100 furol seconds at 210 F. until the softening point is about 200 to 220 F., and a mineral lubricating oil sufficient to characterize the lubricant with a viscosity of about 140 to 200 furol seconds at 250 F.

Lubricating Compositions Containing 2-Benzothiazyl - N,N - Dialkyldithio-Carbamates, Patent No. 2,785,129 (W. S. Quimby and J. M. Robertson, assignors to The Texas Co.) A hydrocarbon lubricating oil containing a corrosion inhibiting amount of a 2-benzothiazyl-N,N-dialkyldithiocarbamate.

Extreme Pressure Lubricant, Patent No. 2,785,130 (T. W. Langer, assignor to The Texas Co.) A lead soap extreme pressure liquid lubricant comprising as the essential ingredients a mineral lubricating oil as the predominating constituent, about 3 to 15% by weight based on the lubricant of lead naphthenate as the sole metal soap in the lubricant, about 1 to 25% by weight of an oil-soluble organic extreme pressure agent selected from the group consisting of chlorinated paraffin wax containing about 35-47% by weight of combined chlorine, sulfo-chlorinated sperm oil containing about 4-6% of combined sulfur and about 4-6% of combined chlorine, and mixtures thereof, and about 0.1 to 3.0% by weight of added glycerine effective to increase the extreme pressure properties of the said lubricant.

Process for Preparing Alkyl Phenol Sulfides, Patent No. 2,785,131 (T. F. O'Connor, assignor to Esso Research & Engineering Co.) In a method for preparing mineral lubricating oil solutions of barium alkyl phenol sulfides, the improvement which comprises reacting alkyl phenol with sulfur chloride in the presence of a saturated acid-treated mineral lubricating oil essentially free of olefinic and aromatic constituents, wherein said lubricating oil is a white oil having a SSU viscosity at 100 F. of about 30 to 100 seconds, and then preparing said barium alkyl phenol sulfide in the resultant white oil solution of alkyl phenol sulfide.

Process for the Extraction of Mineral Oil Contaminants from Synthetic Lubricants, Patent No. 2,785,194 (M. F. Hoare, assignor to Esso Research & Engineering Co.) A process for the removal of mineral oil from synthetic ester lubricating oil compositions which have become contaminated with mineral oil wherein said synthetic oils are polycarboxylic acid esters consisting of hydrocarbon chains joined by ester groups, said chains being selected from the group consisting of aliphatic hydrocarbon chains and aliphatic hydrocarbon chains containing ether linkages therein, which comprises the steps of contacting the said contaminated synthetic lubricating oil with a substantially anhydrous aliphatic saturated hydrocarbon monocarboxylic acid containing from 1 to 6 carbon atoms per molecule, whereby the synthetic ester lubricating oil dissolves in said acid and forms therewith a phase separate from the mineral oil, separating the two phases so formed and extracting the synthetic lubricating oil from said acid phase.

Evaluation of Oils

(Continued from p. 103)

and water test, increased heating and centrifuging time was necessary with some oils. Investigation of the steam emulsion numbers of these oils showed that SE numbers in the 300 to 400 range were unacceptable from an economic point of view. Between 200 and 300, oils could be used, but with great difficulty. From 200 seconds down, it was noted that better oil could more easily and consistently be obtained, the optimum being obtained at SE numbers below 120.

CONCLUSION

The testing procedures as outlined in this paper are not necessarily the best or the most accurate ways of determining lubricant suitability. However, they have been used successfully and economically to examine many commercial and laboratory products and to predict the performance of these materials with a fair degree of success.

BIBLIOGRAPHY

G. Sachs and K. R. Van Horn "Practical Metallurgy", American Society for Metals.

Standardization of Plant Lubricants

(Continued from p. 97)

from ± 5 to ± 25 seconds. Sometimes such substitutions were not possible because of characteristics other than viscosity, such as extremely low pour point, dielectric strength, anti-foam, etc. These oils were classified as special purpose lubricants. Based on the information we had assembled, four charts were developed.

1. Reference chart for the selection of straight mineral oils.
2. Selection chart for non-corrosion and/or oxidation-inhibited oils.
3. Selection chart for extreme-pressure oils.
4. Selection chart for greases.

An example of the Selection Chart is shown in Figure 2.

The publication of these charts indicated to the lubrication personnel what oils we considered should satisfy all our applications. However, we had to conduct plant trials in many cases to effect the proposed changes. These charts are revised from time to time to reflect changes in our operations as well as the use of improved lubricants made available by our suppliers.

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SUMMARY

The procedure outlined is one method used in standardizing plant lubrication. There are no hard and fast rules for accomplishing a good lubrication program. The important consideration is to have a definite approach in mind and time limits on the completion of each phase. While considerable stress has been placed on the actual reduction in the cost of lubricants, it must be remembered that the intangible savings are more than likely more important reasons for such a program. However, in explaining the advantages of a lubrication program to management, cost figures are readily obtainable and do indicate part of the benefit to be expected.

A good lubrication program, therefore, should contain the following requirements:

1. A coordinator or group of coordinators with the proper control over lubrication personnel.
2. An educational program for lubrication supervision as well as those dispensing the lubricants.
3. A good preventive maintenance program.
4. Lubricant specifications.
5. Continued vigil and pursuit of better lubricants and lubrication practices.

The initiation of such a program may take many months. The expected results may be slow in appearing; however, they will appear and with them the full realization that they are worth the work and time expended.

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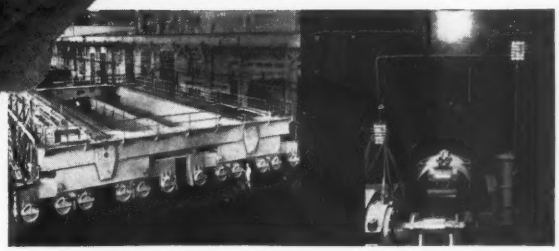

One of the three giant Morgan ladle cranes prior to being outfitted with Trabon Centralized Lubricating Systems. Note immensity of the lubricating job if it had to be done manually.

Trabon automatic lubricant pump, two feeders and lubricant lines. Note cam in right foreground which automatically starts the lubricating cycle while crane is in operation. No auxiliary machinery is necessary. Trabon is easily and economically installed on hydraulic, mechanical, electric motorized and pneumatic equipment.

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